The FEBRUARY, 1933

FEB 1 1933

Heating · Piping and Air Conditioning

IN THIS ISSUE

Cooling Water for Air Conditioning or Industrial Process Uses with Steam Jet Refrigeration 77

Hot-Water Heating for Skyscrapers—Sixty Wall Tower, New York City 82

Piping a Sand-Water Mix 87

Designing Copper Gaskets for Hydraulic Piping 88

Central Steam Plant and Pipe Tunnels Keep Heating Costs Down at U. of C. 90

Foundry Saves $6,000 Annually in Exhaust-Fan Bearing Maintenance 94

On the Job 95

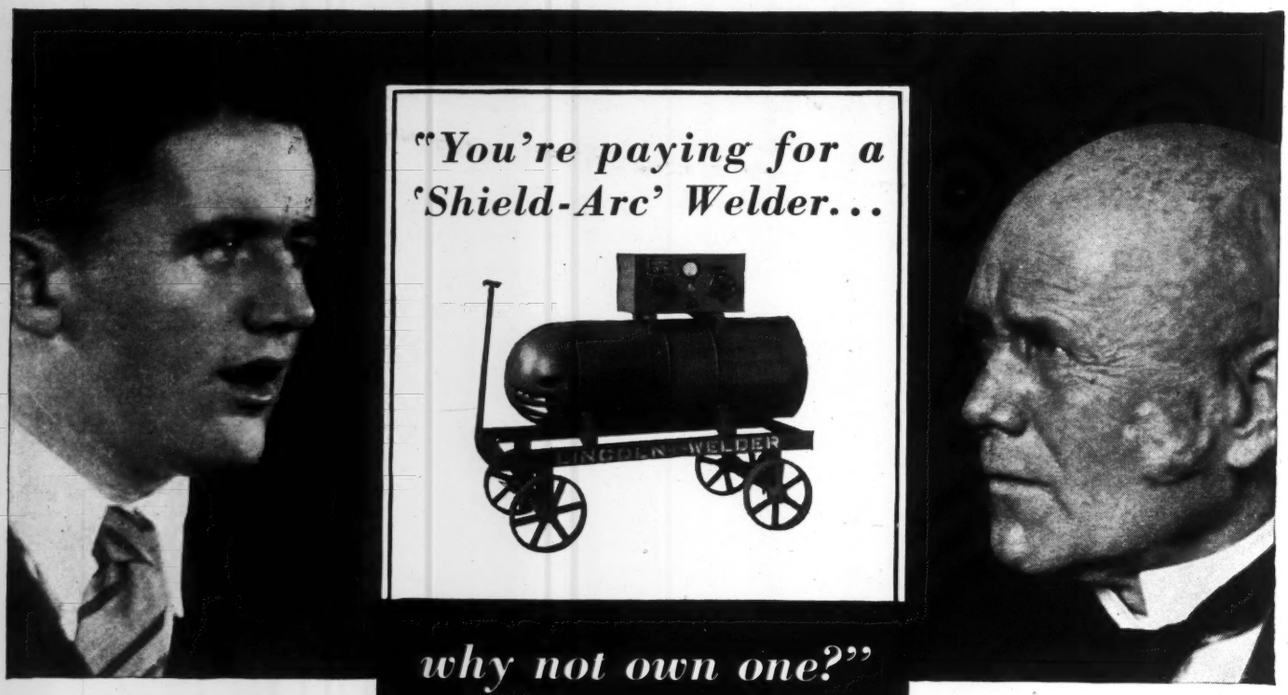
Controlling Motors for . . . Temperature Control Systems . . . Compressors . . . Oil Burners . . . Stokers 100

Units Condition Concrete Testing Laboratory 104

Interesting Equipment Developments 106

Bernoulli's Theorem—What It Means 111

« « *and the* Journal Section *of the* American Society of Heating and Ventilating Engineers



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Heating · Piping and Air Conditioning

THE EDITOR'S PAGE

MODERNIZATION of heating, piping and air conditioning systems in industrial plants, office buildings, hotels, hospitals, schools and other large structures will not be neglected in future issues—nor is it in this one—of HEATING, PIPING AND AIR CONDITIONING despite last month's number being devoted entirely to the theme "Modernize—For Savings and Profit." To us, modernization means to replace where replacements are necessary, add where additions will improve, change where changes will benefit, repair where repairs will suffice. Modernization of this kind is good engineering and a part of our editorial program for future months is to report how the application of these principles will benefit plants and buildings.

FOR INSTANCE, A. W. Moulder—member of our Board of Consulting & Contributing Editors—is now at work on an article for early publication which will give the complete story on the unit-type air conditioning equipment for a Dallas, Texas, department store. Air conditioning for enterprises which depend on attracting the public is distinctly a paying proposition. Particularly is this true in the southern states; readers will recall the description of the central system for air conditioning the D. H. Holmes Co. department store, New Orleans, in last month's issue. Air conditioning large stores and similar buildings seems to be comparatively active at the present time.

A NUMBER of other valuable articles are on the boards for early publication; there's R. C. Doremus' (also a Contributing Editor) paper on modernization of industrial refrigeration piping, containing tables and curves essential to a thorough check-up. T. K. Sherwood and E. W. Comings of M. I. T.—newcomers to our pages—write on the research on industrial drying in progress at that institution, presenting graphical data you will want to have. Articles on heating—an essential service in every building—will be numerous and worthwhile. Steam and process and the numerous other varieties of piping found in the modern industrial plant will not be neglected by our contributors.

DON'T MISS Walter L. Fleisher's article on steam-jet refrigeration—much talked-of these days—starting in *this* issue; or T. S. Tenney's comprehensive description of hot-water heating for a 67-story skyscraper and how this type of system is applied to tall buildings; or "On the Job," which includes three short practical articles this month. Better yet, don't miss any of this month's number!

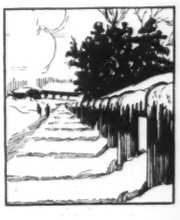

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TABLE OF CONTENTS—FEBRUARY, 1933

Cooling Water for Air Conditioning or Industrial Process Uses with Steam Jet Refrigeration, by Walter L. Fleisher 77

Hot-Water Heating for Skyscrapers—Sixty Wall Tower, New York City, by T. S. Tenney 82

Piping a Sand-Water Mix, by C. L. Mantell 87

Designing Copper Gaskets for Hydraulic Piping 88

Central Steam Plant and Pipe Tunnels Keep Heating Cost Down at U. of C., by Charles Levin 90

Foundry Saves $6,000 Annually in Exhaust-Fan Bearing Maintenance, by Francis A. Westbrook 94

On the Job 95

Controlling Motors for . . . Temperature Control Systems . . . Compressors . . . Oil Burners . . . Stokers, by Samuel R. Lewis 100

Units Condition Concrete Testing Laboratory 104

Interesting Equipment Developments 106

Bernoulli's Theorem—What It Means, by J. C. Reed and E. E. Ambrosius 111

● ● ● ● ● ●

Journal Section of the *American Society of Heating and Ventilating Engineers*

Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo 115

Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich 127

Bulletin Section 132

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The February, 1933

Vol. V, No. 2

Heating · Piping and Air Conditioning

Cooling Water for Air Conditioning or Industrial Process Uses

with Steam Jet Refrigeration

Historically, steam-jet refrigeration is not new but its particular ability to furnish cooled water at temperatures required in air conditioning and improvements in the design of thermo-compressors make it a much-talked of subject today.

The present article is most comprehensive and is devoted to the use of steam-jet refrigeration equipment for supplying cold water for industrial process work and industrial air conditioning. In this issue, the fundamentals are explained and steam and condenser water requirements are discussed. Next month, examples of industrial applications of this type of refrigeration will be considered.

In addition to these industrial applications, vacuum refrigeration has been used for comfort air conditioning in restaurants, trains, etc., for cooling drinking water and for the direct cooling of tomato juice, milk and other liquids, the liquid to be cooled being itself the refrigerant.

• • • By Walter L. Fleisher*

WHEN steam at sufficient pressure is ejected from a properly designed nozzle into a secondary compression chamber, the velocity of ejection entrains the vapors from a connected evaporating vessel, and creates sufficient additional kinetic energy to maintain the vacuum created and discharge the gases or vapors into a chamber where these vapors may be condensed and carried off. This is the cycle of refrigeration that is accomplished with the steam-ejector refrigerating apparatus, and one that can simply cool water for air conditioning or industrial process use or even — under favorable conditions—produce ice.

When steam is generated against higher pressure than the atmosphere, it is generated at higher temperatures than 212 F, and when steam is generated at pressures less than the atmosphere, it is at temperatures less than 212 F. We are so unconscious of the weight of air and moisture under which we live that we do not usually realize this tremendous pressure. This is because we are never conscious of those things to which we are thoroughly accustomed. At sea level there is about 15 lb of air pressure on every square inch of surface, and whether we breathe, or boil water, or do any work, we are always doing these things against that much resistance.

Steam and Its Temperatures

The work that must be done is translated either into a change of state or a change of heat, or both, and these two changes represent basically, the equilibrium of the universe. Consider the case of water in an open vessel, add what heat you will, it never rises to a temperature higher than 212 F. But sooner or later, if the heating process is continued, a vapor—steam—is given off, the extra heat added going into the process of changing the water from a liquid to a vapor or gas, the amount of heat necessary to create a change of state being many times greater than the amount necessary to change the temperature. If the heating is continued for a sufficient time, all of the water will disappear, the vapor being dissipated in the surrounding atmosphere but always, as long as any water remains in the vessel, it will be at 212 F. And so boiling or steam-making and 212 F have become synonomous to all of us.

Few of us stop to realize that all of this has taken place under a pressure of 15 lb per sq in. Now, close the open vessel and, as the heat is added, the thermometer will rise above 212 F before much steam or vapor has come off the surface of the water. At a pressure of 2 lb gage, the temperature of boiling is 220 F, and

*Consulting Engineer, New York City. Member of Board of Consulting and Contributing Editors.

at 100 lb gage (a very common pressure in industry), 338 F.

At 1/12 Lb Absolute, Water Boils at 32 F

Similarly, if the atmospheric pressure is removed, the temperature of boiling is lowered. Remove ½ of the 15 lb and water begins to boil at 180 F; at ¼ of 15 lb water boils at about 150 F; at ¼ lb absolute, between 50 and 60 F; and at ⅛ lb absolute, at 40 F; at 1/12 lb absolute, 32 F. This is the ordinary phenomenon of water, overlooked so often by both our engineering and lay minds, but which is responsible for our present interest in so-called vapor vacuum refrigeration.

How a Steam Ejector Operates

Fig. 1 shows, diagrammatically, the operation of a simple steam-ejector system. *A* is the nozzle fed from a steam header with steam at any predetermined pressure. This nozzle can be of any of the well-known velocity-type nozzles, to give a velocity at the orifice of about 4,000 fps. The kinetic energy which is developed delivers the steam in a cone, which cone is compressed at *B* in order to seal the venturi tube *C* to prevent back action which might break the vacuum. This kinetic energy or velocity draws the air from the evaporator chamber *D* and, if no air exists in this chamber, draws any other gases or vapors which may be in this chamber up to the ultimate entraining capacity of the cone of steam from the face of the nozzle to the point of compression.

This kinetic energy which is developed does two things: First, it removes the air, if any exists, and, secondly, it removes vapor. And it removes the vapor at a pressure and corresponding temperature proportional to its total energy. It removes the vapor from evaporator *D* at a temperature corresponding to the absolute pressure that its own energy creates in the suction connection, and less work is required and the work is more effi-

SATURATED STEAM: TEMPERATURE TABLE

Temp. Fahr.	Abs. Press. Lb./Sq. In.	Specific Volume Sat. Liquid	Evap.	Sat. Vapor	Total Heat Sat. Liquid	Evap.	Sat. Vapor	Entropy Sat. Liquid	Evap.	Sat. Vapor
32°	0.0887	0.01602	3301	3301	0.00	1073.4	1073.4	0.0000	2.1834	2.1834
40	0.1217	0.01602	2445	2445	8.05	1069.1	1077.1	0.0162	2.1397	2.1559
50	0.1780	0.01602	1704.8	1704.8	18.06	1063.6	1081.7	0.0361	2.0870	2.1231
60	0.2561	0.01603	1208.0	1208.0	28.05	1058.2	1086.2	0.0555	2.0364	2.0919
70	0.3628	0.01605	869.0	869.0	38.03	1052.7	1090.8	0.0746	1.9877	2.0622
80	0.5067	0.01607	633.8	633.8	48.00	1047.3	1095.3	0.0932	1.9407	2.0340
90	0.6980	0.01610	468.5	468.5	58.00	1041.8	1099.8	0.1115	1.8955	2.0070
100	0.9487	0.01613	350.8	350.8	67.93	1036.3	1104.2	0.1295	1.8518	1.9813
110	1.274	0.01616	265.8	265.8	77.89	1030.8	1108.6	0.1472	1.8095	1.9567
120	1.692	0.01620	203.5	203.6	87.88	1025.1	1113.0	0.1646	1.7685	1.9331
130	2.221	0.01625	157.62	157.64	97.86	1019.4	1117.2	0.1816	1.7288	1.9104
140	2.887	0.01629	123.20	123.22	107.84	1013.6	1121.4	0.1984	1.6903	1.8887
150	3.716	0.01634	97.21	97.23	117.84	1007.7	1125.5	0.2149	1.6530	1.8679
160	4.739	0.01639	77.38	77.40	127.85	1001.8	1129.6	0.2312	1.6168	1.8479
170	5.990	0.01645	62.13	62.14	137.85	995.8	1133.7	0.2472	1.5816	1.8288
180	7.510	0.01650	50.26	50.28	147.87	989.8	1137.7	0.2629	1.5475	1.8105
190	9.336	0.01656	40.99	41.01	157.89	983.8	1141.7	0.2785	1.5144	1.7929
200	11.525	0.01663	33.65	33.67	167.94	977.7	1145.6	0.2938	1.4822	1.7760
210	14.123	0.01669	27.82	27.83	177.99	971.5	1149.5	0.3089	1.4508	1.7597
220	17.188	0.01676	23.14	23.16	188.06	965.1	1153.1	0.3238	1.4200	1.7439
230	20.78	0.01683	19.371	19.388	198.15	958.6	1156.7	0.3386	1.3900	1.7286
240	24.97	0.01690	16.307	16.324	208.26	952.0	1160.2	0.3532	1.3607	1.7138
250	29.82	0.01698	13.807	13.824	218.39	945.2	1163.6	0.3675	1.3320	1.6995
260	35.43	0.01706	11.745	11.762	228.55	938.4	1166.9	0.3817	1.3040	1.6856
270	41.85	0.01714	10.044	10.061	238.74	931.4	1170.1	0.3958	1.2765	1.6723
280	49.20	0.01723	8.626	8.644	248.95	924.2	1173.2	0.4097	1.2496	1.6593
290	57.55	0.01732	7.442	7.459	259.20	917.0	1176.2	0.4234	1.2233	1.6467
300	67.01	0.01742	6.446	6.464	269.48	909.6	1179.1	0.4370	1.1974	1.6345
310	77.68	0.01752	5.606	5.623	279.80	902.1	1181.9	0.4505	1.1721	1.6226
320	89.65	0.01762	4.892	4.910	290.17	894.4	1184.6	0.4638	1.1472	1.6110
330	103.03	0.01773	4.285	4.303	300.59	886.5	1187.1	0.4770	1.1227	1.5998
340	117.99	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887

Abstracted from "Steam Tables and the Mollier Diagram," by Prof. J. H. Keenan, 1930 Edition, by permission of the publisher, *The American Society of Mechanical Engineers.*

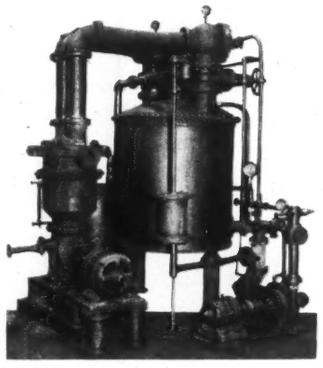
Brief History of Steam-Jet Refrigeration

Historically, vacuum refrigeration using water as the refrigerant is old. Experiments with vacuum pumps to produce low temperature evaporation started almost with the first laboratory development of suction pumps. Boyle, so prominent in chemistry and physics, in the 17th century (1627-1691) made one of the better types of vacuum pumps. His name is mentioned here because the laws he enunciated are basic today in our study of the thermodynamics of heat relationships.

In 1775, a Dr. William Cullen developed a so-called vacuum refrigerating machine using a pump alone, and in 1810, Sir John Leslie made ice in commercial quantities with a pump and a concentrated sulphuric acid adsorber.

John Patten, however, seemingly was the first person with enough enthusiasm to get capital to construct a commercial ice-making plant using a vacuum system working at low enough absolute pressures to develop ice. This plant used a steam-driven vacuum pump and heavy H_2SO_4 as a partial adsorber, and was described in *Ice and Refrigeration,* February, 1904. Unfortunately, a notice of a receivership and liabilities of $200,000 are quoted for the same plant in a later issue.

The pioneer in this country to employ a high-velocity jet to obtain a vacuum sufficiently low to evaporate water for ice-making purposes developed the vacuum refrigeration machine illustrated here. The genesis of the idea employed came from a Frenchman named Leblanc, and was described in 1910 in an issue of *L'Industrie Frigorific* by C. De Chessin Cherchevosky. This method was taken up immediately, and the first plant was erected at Bethune, France. This Westinghouse-Leblanc system is the predecessor of our present development.



A pioneer vacuum refrigeration machine

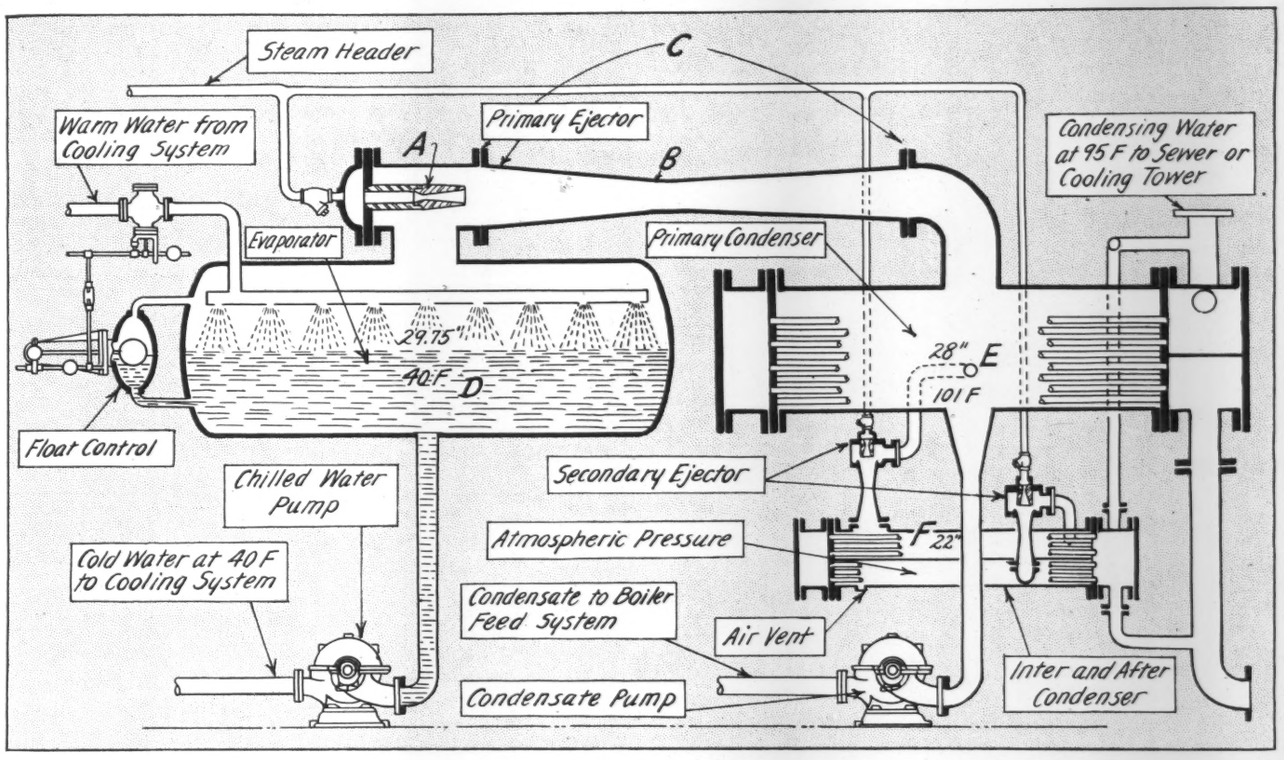


Fig. 1—Diagram illustrating the operation of a steam-ejector refrigeration system; operation is explained in the text

ciently done if the amount of vapor to be removed is obtained from the maximum amount of surface. (This same idea has been used in drying.)

Each Pound of Vapor Removes 1000 Btu

For each pound of vapor which the high-velocity jet can dissociate and remove, approximately 1,000 heat units are removed at the same time, and for each pound of vapor that is removed at the conditions indicated on the sketch, approximately 3,000 cu ft of vapor is created. Each pound of vapor removed therefore has the capacity to cool the *remaining water* by the 1,000 heat units removed—in other words, 100 lb of water 10 degrees, or 50 lb of water 20 degrees, etc, at the temperature corresponding to the absolute pressure.

Pressure and Temperature in the Condenser

As the ejector must maintain a velocity, must entrain and remove a certain weight of vapor, and must do it against the resistance existing beyond the restricted throat of the thermal compressor, it is essential that the pressure on the condenser side of the compressor should be as low as possible. But that absolute pressure in the condenser will also correspond to a condensing temperature which must be within the range of the available condensing medium, usually water. Therefore, in the primary condenser marked *E*, if the absolute pressure is less than 28 inches of vacuum (about 1 lb absolute pressure) that is 28.5 or 29 (about ¾ or ½ lb pressure), although the work to be done by the jet would be less, the temperature of the available condensing medium would be probably so close to the condensing temperature in the condenser *E* that either no condensation would take place, or very large volumes of water would be required.

This requires slightly more explanation. At 28 inches of vacuum the temperature of evaporation and, consequently, condensation is 101 F. Therefore, if the available water is at 70 F there is a 31-F differential between the condensate and the water, so each gallon of condensing water would remove 8⅓×31 Btu from the vapor, or 257 Btu. Or 4 gallons of this water would condense a pound of steam (heat content 1,000 Btu) into water which could be removed with a pump or through a barometric leg. If 30 lb per hr of steam were required to produce a ton of refrigeration, the heat to be removed in the condenser would be 30 times the total heat, or approximately 33,500 Btu, plus the vapor developed in the cooling process, or another 12,000 Btu, a total of 45,500 Btu per hr. If each gallon of water at the conditions specified has a cooling effect of 257 Btu, then for each ton of refrigeration approximately 185 gallons of water per ton per hour would be required, or 3 gpm.

Consider a second example: In order to save mechanical work, the condition in the condenser was maintained at 28½ inches of vacuum (instead of at 28 inches as above); the condensation temperature would be 92 F. With 70 F water, there would be a 22-F differentials (as compared to 31 F in the former example) and only about ⅔ of the work that was obtained with the higher condenser pressure would be done by each gallon of condenser water (22/31). The lower differential between the evaporator and condenser pressures, however, would cut the steam consumption per ton sufficiently so that the *actual* number of gallons required for condenser purposes would be about the same.

(If condenser water was available only at tempera-

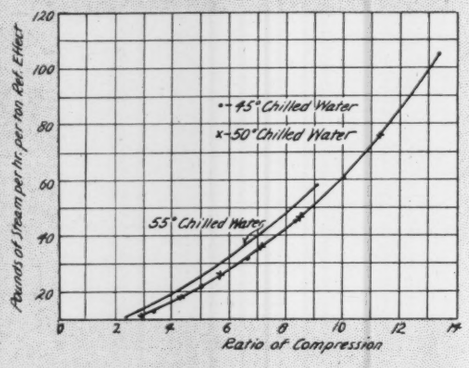


Fig. 2—Variation in pounds of steam per hour per ton of refrigeration for different ratios of compression, with an initial steam pressure of 100 lb gage

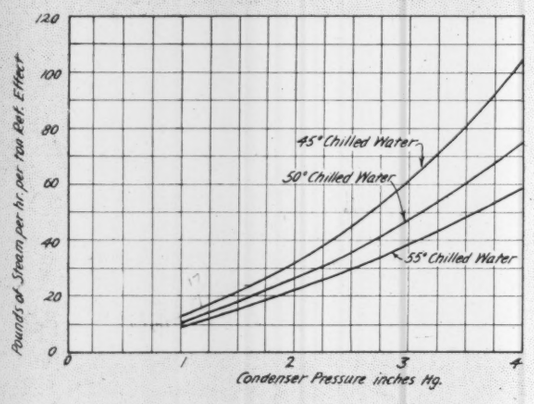


Fig. 3—Pounds of steam per hour to produce one ton of refrigeration for different condenser pressures

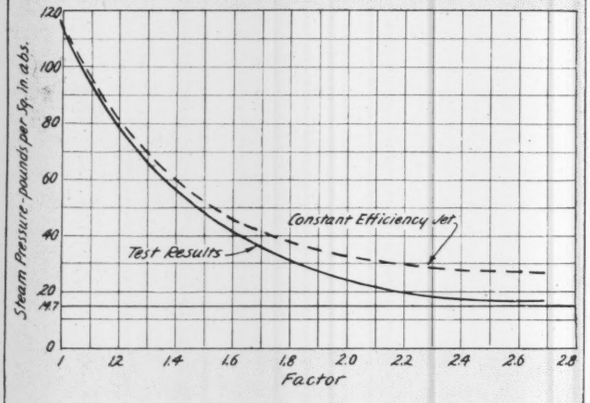


Fig. 4—Relative amount of steam required to produce a ton of refrigeration effect with different initial steam pressures

tures of 85 F, very little condensation would be possible at 28½ inches vacuum. But the differential of 16 F at 28 inches vacuum would be ample for condensing purposes and indicate a marked advantage of the steam-ejector type of refrigeration. This particular example is cited to show the advantage of this type of refrigeration in warm climates or where the available supply of water is at a high temperature. This is particularly true in many cities in the South, in tropical climates, and on ships traveling through tropical or Southern waters.)

Removal of Air and Inert Gases

In Fig. 1 the secondary ejector F is for removing the entrapped air and the inert gases which are carried over in the steam and in the water vapor. These gases have a great tendency to lower the efficiency of the primary condenser and as they are not condensable they must be removed from the primary condenser as they develop. This is readily done with a secondary ejector operating between the absolute minus pressure of 28 inches in the primary condenser, and possibly 22 inches in the secondary condenser. This second condenser F is often an integral part of the primary condenser.

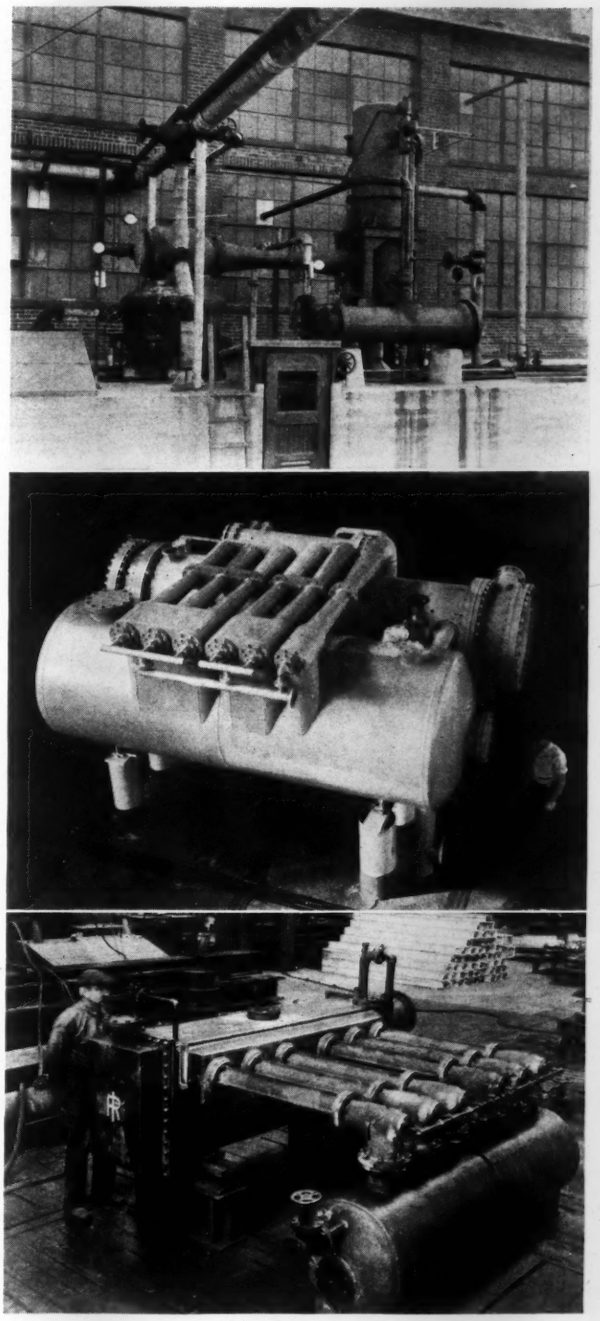
Occasionally the entrapped air is removed by a water jet but where the water temperatures are high or approach closely the temperature of the condenser water, the efficiency is very low and an air pump or a steam ejector is the simplest means of removal. If a steam ejector is used on the secondary condenser, a third ejector and a third condenser must be employed to remove the condensation or vapor from the secondary condenser, so that it may be condensed at atmospheric pressure and discharged from the system.

Use of Warm Water from Primary Condenser

The steam or vapor which is condensed in the primary condenser E must be removed against a vacuum of 28 inches. This water—at about 100 F—can be re-used in the boiler feed system or discharged, as the economics of the situation dictate. On one job, both the cooled water and this warm water are used in the plant process.

It is often advisable to use some of this water for make-up purposes in the evaporator chamber. To the lay mind this might seem to be a very uneconomical thing to do. But, as this water is free of entrained or entrapped air, and as only the heat of the liquid down to 50 F has to be removed, it requires the removal of 50 Btu per lb of water used. As the ordinary water which otherwise would be used would probably be at 70 F in the summer, actually a net of only 30 Btu would be wasted in the process, as against a total of 1,050 Btu required for ordinary evaporation.

As the secondary and tertiary (second and third) ejectors use about 10 per cent of the total amount of steam required for producing the refrigeration effect, the 3 per cent additional work ($\frac{30}{1050} = 3\%$ approx.) to remove the heat from the condenser water would be more than made up by the saving in power necessary to remove entrapped air and vapor, or the con-



Top—Vacuum refrigeration equipment for supplying industrial cold water, located outdoors where space was not a factor. Center—One of the largest units constructed to date, with a capacity of 180 tons when cooling water to 45 F with 125-lb steam. Considerable of the control is obtained with the multiple jets. Bottom—An industrial unit with multiple jets for close control

denser water required would be lessened sufficiently to take care of this inefficiency.

Removal of Cooled Water

The pump shown on the evaporator, working under high vacuum, is used to remove the cooled water for use in its particular device, either an interchanger such as a radiator, or an air washer. Although much more accurate results can be obtained in a washer, nevertheless the circulation of this water through an interchanger has certain advantages in that, except for slight leakage, no air enters the system. In a washer large amounts of entrapped air are picked up which add to the work and required size of the air removal apparatus.

Steam Consumption of Ejector Systems

The condenser pressure has a very marked effect on the steam consumption in the vacuum steam jet refrigeration cycle. Like any other compression system, the ratio of compression divided by the mechanical efficiency is the determining factor in estimating the horsepower. In the steam ejector refrigerating system the ratio of compression between the evaporator and the condenser, divided by the mechanical efficiency is the determining factor in arriving at the amount of steam required.

Fig. 2 indicates the variation in pounds of steam per hour per ton of refrigeration with different ratios of compression, with an initial steam pressure of 100 lb gage. Theoretically, these curves should coincide; the variation in their position is probably due to inaccuracies in the tests or in design of the nozzles.

The amount of steam required per hour per ton of refrigeration, with 100 lb gage initial pressure, is indicated in Fig. 3. This shows, with different ratios of compression, the relative amount of steam required to maintain different temperatures in the evaporator, and Fig. 4 indicates the relative amount of steam required to produce a ton of refrigeration effect with different initial steam pressures.

As an example, from Fig. 4, with 115 lb absolute, if 30 lb of steam were required to produce a ton of refrigeration effect per hour, then with 40 lb absolute or 25-lb gage pressure, 1.6 times that much would be required, or 48 lb of steam per ton of refrigeration. As the pressure drops to approximately 5 lb it can be seen that 64½ lb of steam would be required. In actual practice, with nozzles as now designed, the steam consumption runs well above 64½ lb at this low pressure.

New Book on Heat Transmission

"Heat Transmission," by Prof. Wm. H. McAdams, has just been published by McGraw-Hill Book Co., Inc., New York City, under the auspices of the committee on heat transmission of the *National Research Council*. It consists of 355 pages, 6 x 9, with 135 figures and 68 tables; price $5.

The section on conduction consists of two chapters, the first dealing with steady conduction of heat, thermal conductivities, the effect of the shape of bodies, and resistances in series and in parallel. In the second chapter, unsteady conduction, as in the heating and cooling of solids, is considered, and problems are solved by the use of charts involving four dimensionless ratios.

The second section treats radiation between solids for a number of important specific cases, radiation from non-luminous and luminous flames, and the general problem of furnace design, where heat is transferred simultaneously by several mechanisms.

The third section consists of seven chapters on dimensional similarity, fluid dynamics, convection, and the measurement of surface temperatures. The remaining chapters deal with heat transfer under the headings of fluids inside pipes, fluids outside pipes, condensation, and evaporation.

By T. S. Tenney*

Hot-Water Heating for Skyscrapers

—Sixty Wall Tower New York City

Fig. 1—The Sixty Wall Tower building. From street level through the 30th floor, the building is 125 ft by 250 ft, where it is set back to 80 ft by 195 ft. The tower starts from the 31st floor with a dimension of 80 ft by 120 ft and is set back at the 63rd to 58 ft by 66 ft. The observation gallery at the extreme top is 30 ft by 40 ft and is 55 ft above the 63rd floor level. From the street level to the light on the topping pole is 950 ft.

Owner, The Henry L. Doherty Co.; Architects, Clinton & Russell, Holton & George; Consulting Mechanical Engineers, Tenney & Ohmes; Heating Contractor, Riggs, Distler Company, Inc.

DISTINCTIVE among the tall buildings of the world is the 67-story Sixty Wall Tower in New York City. All floors above the first are heated by a zoned hot-water heating system, with radiation of the convector type consisting of finned tubes expanded into headers. The first floor and below is heated with low-pressure steam, ordinary cast-iron radiation being utilized. The hot-water system is divided into four zones each of approximately fifteen floors and a fifth zone from the 63rd floor to and including the observation tower. With this arrangement the radiation on the lower floor of each zone is subjected to the maximum pressure of 115 lb per sq in; this is well within the test pressure limit, as the radiators used were tested hydrostatically at 750 lb per sq in.

The piping is seamless steel with a wall thickness slightly heavier than standard pipe. It was subjected to a test pressure 40 per cent in excess to that which standard lap-welded pipe is ordinarily subjected. The joints of all main line piping are welded, except at the circulating and heating units, where heavy cast-steel flanges are used. The fittings on the rising lines of each zone which feed the heating surfaces are of forged steel, drilled and tapped, and are capable of withstanding a pressure in excess of the bursting strength of the pipe.

The central heating units for the two upper zones, representing the maximum pressure, are subjected to a pressure of 400 lb per sq in., and the larger units for the two lower zones to 300 lb per sq in. and 140 lb per sq in., the latter being the pressure at the base of the lower zone. All these pressures are within the safe limits of the materials and apparatus used.

*Tenney & Ohmes, Consulting Engineers, New York City. Member of Board of Consulting and Contributing Editors.

Street Steam Drives Turbines; Exhaust Heats Water

Steam for heating, ventilating, power, and for domestic hot-water heating is supplied by the district steam company. A distributing header is provided, divided by valve, for the heating and ventilating load (winter service) and for the domestic hot-water load (constant service).

Steam from the street mains is furnished at 135 lb; from the distributing header branches are taken for the turbine-driven house water-supply pumps and hot-water heating pumps, and for a reciprocating engine-driven air compressor. Exhaust steam from these units is discharged into a low-pressure steam distributing system and used for the heating and ventilating services.

The exhaust steam output represents about 40 per cent of the average low-pressure steam demand and it is necessary to reduce 60 per cent from the street pressure of 135 lb to approximately 2 lb. The use of steam drive secures the power for water circulation and pumping at a cost represented only by the slight heat loss in the steam-driven units and the exhaust steam piping. Reducing valves for reducing the required amount of high-pressure steam to the exhaust-steam pressure are installed in multiple to meet the varying low-pressure steam demand.

The water of condensation from the heating, ventilating and domestic hot-water units is returned to open tank receivers, from where it is pumped to the sewer after passing through economizers, where heat in the condensate is absorbed by the cold-water supply to the domestic hot-water heaters.

Six Heating Zones

The natural sub-division into heating zones was a division of the base and tower each into two zones (four main zones for the office floors) with a separate zone for the first floor and below, and a separate zone above the 63rd floor, the latter covering a special service section.

This zone sub-division is as follows:

Zone *S*—From cellar to and including the 1st floor.
Zone *A*—From 2nd floor to and including the 15th floor.
Zone *B*—From 16th floor to and including the 30th floor.
Zone *C*—From 31st floor to and including the 46th floor.
Zone *D*—From 47th floor to and including the 62nd floor.
Zone *E*—From 63rd floor to and including the observation tower.

Zone *S* is heated by low-pressure steam and is designed as an open return line system, discharging the condensate by gravity to the main open receiver. Zone *S* includes the main entrances to the basement and first floor. With the general public and open spaces, it was deemed advisable not to connect this section with the water heated Zone *A*, as it would, with its varying heat demand, unbalance the centralized water heating temperature control for this zone.

The heating system in Sixty Wall Tower, 67-story skyscraper, is a good example of changes in method made possible by equipment developments. Increase of working pressures in central station work has made available new apparatus and materials. In the case of hot-water heating this has removed former limiting heights; with usual zone subdivisions, hot-water heating can be applied today to large buildings of considerable height.

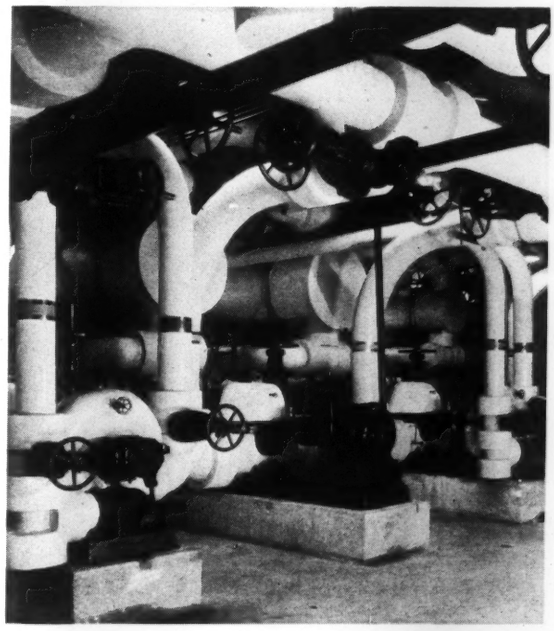


Fig. 2—View of the equipment room showing the hot-water heating units. The pumps of the general service units are turbine driven; auxiliary units have motor drives

Top Zone Heated by Humidified Warm Air

Zone *E* is heated by warm air, the air being taken from the outside or recirculated within the zone. The air is circulated by a motor-driven centrifugal fan, arranged with mat air filters, air tempering and heating stacks for hot-water and a hot-water coil air-moistening pan for adding moisture to the air as required during the heating season. The air is carried to the base of the flues for each room by a double system of air ducts, one conveying air at room temperature and one with air at the heating temperature. A double damper is provided in each connection, the dampers being controlled by room thermostats which actuate the dampers and maintain an even temperature in the rooms through mixing of the tempered and heated air.

Observation Gallery Can Be Cooled

Provision is made at the central air-supply system for cooling during the warm weather by cooling the water normally circulated for heating. The ob-

servation gallery consists of a large glass area which may require cooling to counteract sun heat. With the use of water for heating, such cooling can be readily secured with the installation of the equipment in the cellar at the point of central control.

Equipment for Four "Office Zones" in Basement

Zones *A*, *B*, *C*, and *D* represent the office areas of the building, each being served by an independent heating and water circulating unit located in the cellar for centralized building temperature control. The capacity of each heating zone is given in Table 1.

Each system is of the two-pipe, down-feed type, designed for a circulating head of 60 ft with a temperature drop of 30 F at the heating units of the four main heating zones.

Fig. 2 shows a section of the central heating station, the three units in the foreground serving Zones *A* and *B*, the center unit being an auxiliary for either Zone *A* or *B*. Each unit consists of a centrifugal pump with heater and interconnecting pipe and control valves. The pumps of the general service units are steam-turbine driven. The pump of the auxiliary unit is electric-motor driven. A similar group of units is provided for Zones *C* and *D*, and independent steam-and electric-driven pump units for Zone *E*.

Table 1—Capacities of the Heating Zones

Floors Served	Water Circulated in Lb Per Hr	Steam Radiation Equivalent at 240 Btu Per Sq Ft		Supply Main Inches	Return Main Inches
		For Heating	For Unit Ventilators		
Zone *S*					
Cellar—1st	Steam	14,000	.....	8	3½
Zone *A*					
2nd—15th	650,000	56,000	29,000	8	6 (two)
Zone *B*					
16th—30th	570,000	37,000	34,000	8	6 (two)
Zone *C*					
31st—46th	200,000	10,000	15,000	6	6
Zone *D*					
47th—62nd	132,000	4,800	12,000	6	6
Zone *E*					
63rd—O. G.	50,000	3,500	.....	3½	3½

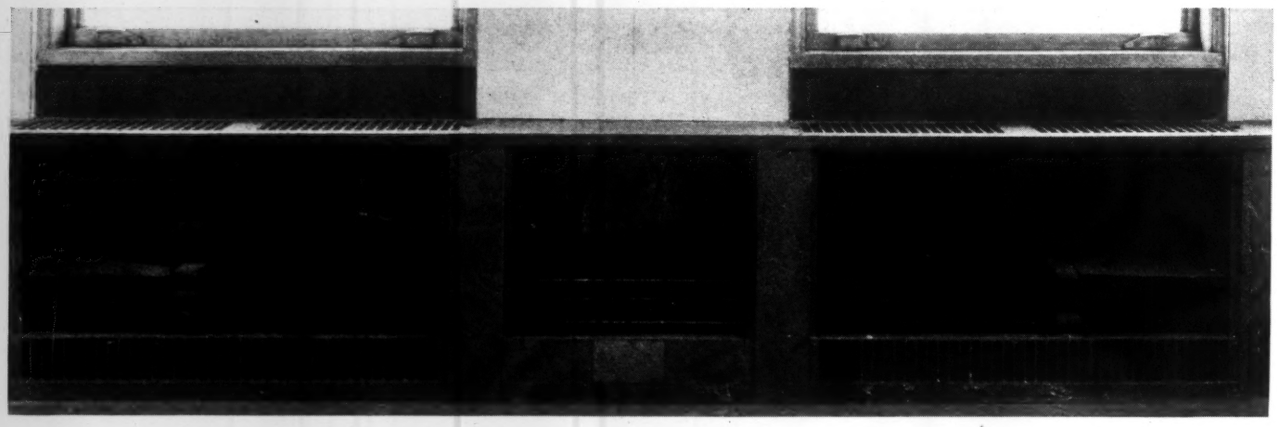
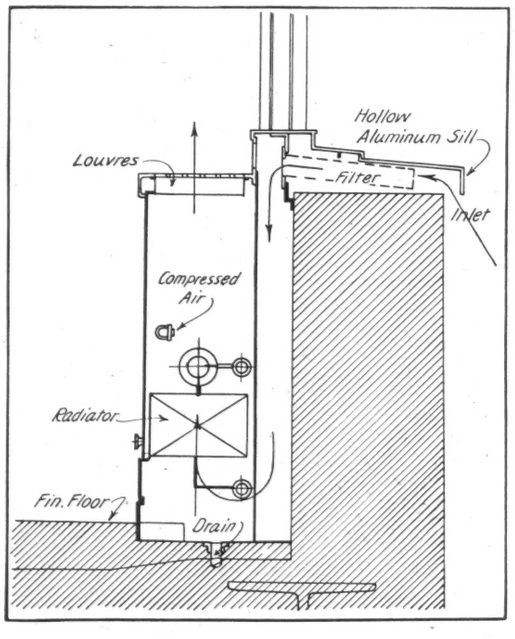


Fig. 3—Arrangement of the heating surfaces is shown above, the enclosure front being removed. The hair-pin bend permits movement of the riser. Below is a cross-section of one of the ventilating units



Flow Mains Divided for Warm and Cold Exposures

The flow main from each heater rises to the top floor of the respective zone where it is divided into two mains, one for north and west (or cold) sides of the building, and one for south and east (or warm) sides. This affords a means of flow regulation to compensate for sun action and wind intensity. From Zones *A* and *B*, the return mains from the cold and warm sides are carried separately to the central point of distribution and each valved for regulation at this point.

From the mains at the top of each zone, risers are carried down in each window bay for supply to the heating surfaces and return risers to and connecting with the return main at the bottom of each zone.

The maximum rising line size is 3 in. in Zones *A* and *B*, the size generally being 2½ in. for *A* and *B* and 2 in. for *C* and *D*. The larger size in *A* and *B* was necessary in the large end bays of three windows, the bays

of only two windows being generally 2½ in. down to 1½ in.

The carrying capacity of the various riser pipe sizes is:

Pipe Size, In.	Equivalent in Steam Radiation, Sq Ft
1½	900
2	1750
2½	2900
3	5000

Flow and return connections for radiators are as follows:

Pipe Size, In.	Equivalent in Steam Radiation, Sq Ft
½	50 and below
¾	51 to 100
1	101 and above

Control valves are provided on each set of rising lines and drain valves and piping at the base of each return riser. Vent lines are taken from the top of each main flow riser and from the ends of each system of flow pipes to and connecting above the water line in the expansion tank. The overflow from each expansion tank terminates over a funnel in the cellar which also acts as the vent, tank drain and tell-tale line.

Heating Surfaces Completely Enclosed

The heating surfaces are completely enclosed, the enclosure extending across the entire width of each window bay. The center portion is a removable panel for the installation of the individual ventilating unit. The grille at the top of the enclosure is of the register type affording a means of local control of heat, if required.

The heating surfaces are of the three-pass type in order to get a large temperature drop through each radiator to reduce the amount of water circulated. They are made up of ¾-in. O.D. half hard copper tubes with 0.035-in. walls and expanded by ferrule into headers at each end. Fins of 0.012-in. copper, with ⅜-in. projection, are metallically connected to the tubes and coated by solder to the tubes, there being seven fins per inch of each tube length.

The surfaces in the base portion contain eleven tubes, staggered and horizontally placed, 4-3-4; in the tower they contain eight tubes, 3-2-3. The water is supplied to a single header at the top, first passing through the top row of tubes to a double header at the opposite end and then through the middle and bottom row by similar headers, and from the bottom to the return risers.

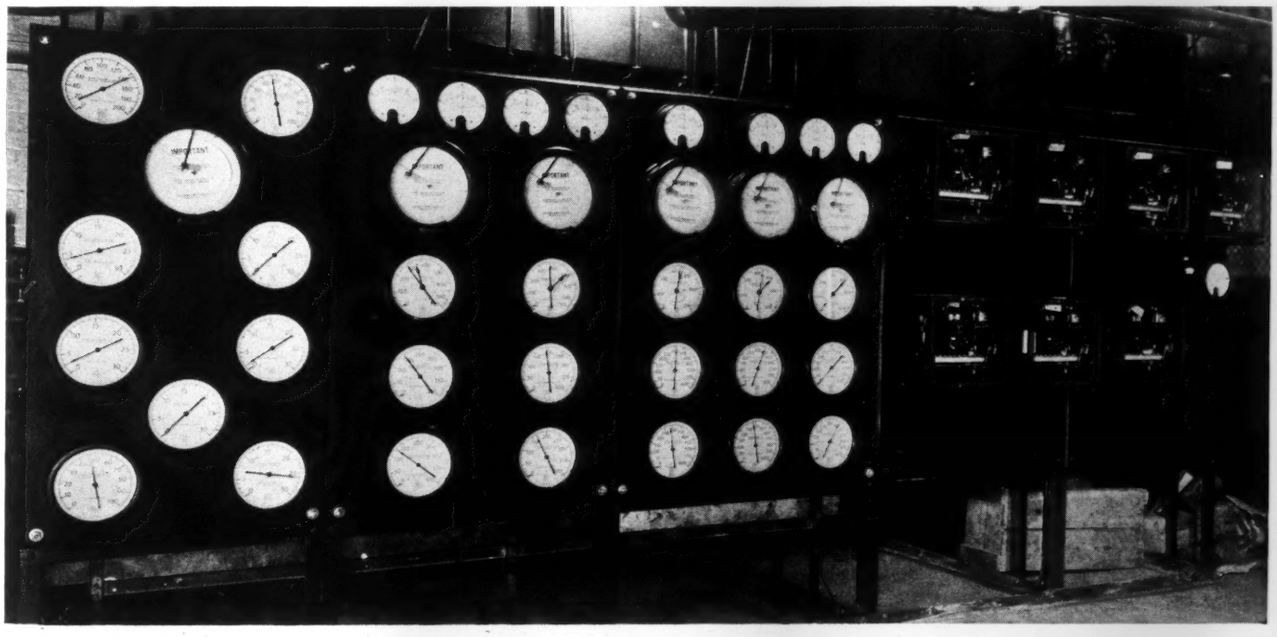
The connections to the surfaces are contained in the enclosures, which are clearly shown in Fig. 3.

Central Control Board Enables Careful Control

Fig. 4 shows the central control board, adjacent to the central heating units; this indicates to the operating engineer the conditions prevailing at all supply sources, as well as showing temperatures throughout the entire building.

At the left of the board are located the gages which indicate the pressures in the various steam and air systems. The lower center portions contain recording thermometers for the water flow temperature of each zone, with gages indicating the circulating differential. Directly above—at the top of the board—are located the direct-reading thermometers which show the temperatures on the cold and warm sides of the building in any of the zones, the elements for these thermometers being located on any floor of the zone recorded and where average temperature conditions prevail. At the right of the board are located the temperature-indicating instruments by which the operator, by the dials to the extreme right, can secure the temperature on any floor throughout the entire height of the building. Wet-

Fig. 4—Located adjacent to the central heating units, the control board—shown below—enables efficient operation of the system. Direct-reading thermometers at the top of the board show temperatures on the cold and warm sides of the building in each of the six building heating zones



and dry-bulb recording can also be taken of the spaces served by apparatus in which provision is made for moisture addition to the air.

These provisions for temperature observation throughout the building and the close temperature regulation that such a heating system permits, makes manipulation of the individual heating surfaces unnecessary. Any regulation to suit individual temperature requirements can be secured, if necessary, by the manipulation of the dampers on the outlets of the radiator enclosures.

Steam-and electric-driven air compressors are installed with a system of compressed-air piping throughout the building, with compressed air outlets at each heating surface for cleaning and for any other purpose to which such service can be applied.

New Developments Make This System Possible

The heating system in Sixty Wall Tower is a good example of changes in method brought about through equipment developments. The increasing of working pressures in central station work has made available in other fields apparatus and materials not procurable in the past; in the case of hot-water heating, this has removed the limiting height for using this method of heating so it can be adapted, with the usual zone subdivision, to large buildings of considerable height.

With hot water, automatic regulation of the temperature by any period or thermostatic method is not usually required, regulation being secured by a gradual increase or decrease of water temperature by hand control of the steam valve to the heater. After once in use, the maximum temperature or movement in the piping is very rarely reached, and when reached, it is gradual and over a considerable period of time. This minimizes piping strains.

The cost of installing hot-water heating in small buildings where gravity is used for circulating the water will often exceed the cost of a steam installation. In large installations, where high-pressure steam is available, water under a high forced circulating head will often cost less than steam for heating. The choice depends of course, on a study of the particular job.

The Sixty Wall Tower building encloses approximately 13,000,000 cu ft and the comparable cost of the heating and ventilating equipment with buildings of like size indicates a saving in cost over steam, when considering that the central ventilating systems are extensive and that the capacity of the heating sysem is designed and heating surfaces provided to serve about 850 individual air supply ventilating units that can be installed at various points on any floor throughout the four office floor zones.

The items representing the greatest saving are the reduction in all pipe sizes and the installation of one heating unit to an outside column bay, thereby reducing the number of individual connected units from 3,650 to 1,660. The increase in heating surface and the apparatus cost for circulating and heating the water must be offset against this. The latter, in this case, represents less than 4 per cent of the total cost of the heating and ventilating equipment.

With the system used in this building the heat losses from the piping, so to speak, can be considered as useful heat at all times, as the water is circulated at a temperature dependent upon the outside temperature or heat demand; with means of close regulation of water temperature, a comfortable room temperature can be maintained within the building and waste through overheating avoided.

This system also affords means of maintaining a partially-heated building; during shut-down periods, such as Sundays and holidays, the minimum amount of heat can be kept in the building, resulting in economy in steam as well as the avoidance of the undesirable high temperatures usual with a completely-closed building.

Data on Steam Consumption Available Next Year

All means have been used to secure a close control of building temperature and if any saving in steam consumption is effected, it will be best shown by comparison with other and similar buildings of like heating and ventilating requirements served with steam from the same outside source. The building thus far has been heated only temporarily; it is hoped that after the present heating season economy in steam consumption will be shown.

Freezing and bursting of a hot-water heating system has sometimes been advanced as a reason against their use. In a building of this type with its vast machinery equipment distributed throughout the building and under constant supervision of a skilled force, the chances of such conditions arising are remote. If freezing is feared, a non-freezing solution can be used as the heat-carrying medium.

Recent Trade Literature

Air Conditioners and Silencers: Campbell Metal Window Corporation, 100 E. 42nd St., New York City, 12-p booklet on units which provide noise elimination, ventilation, cleaning, humidification, cooling, and dehumidification in individual rooms in existing or new buildings.

Alloys: The International Nickel Company, Inc., 67 Wall St., New York City. 8-p technical bulletin on the engineering properties of monel metal.

Convection Heaters: The Hart, Cooley, Highton Company, New Britain, Conn. 8-p catalog of concealed and cabinet convection heaters.

Dust Control: Pangborn Corporation, Hagerstown, Md. 4-p. folder on industrial dust control.

Expansion Joints: Yarnall-Waring Company, Chestnut Hill, Philadelphia, Pa. 16-p bulletin describing and giving dimensions, prices and applications of cylinder-guided gland-packed and gun-packed expansion joints.

Flow Meters and Indicators: Schutte & Koerting Co., Philadelphia, Pa. 8-p bulletin on instruments to show rate of flow of water, air, oil, gas, chemicals and other fluids, both liquid and gaseous, high or low pressure; and flow indicators and lookboxes.

Gages: The Federal Gauge Company (Division of Mercoid Corporation) 4201 Belmont Ave., Chicago, Ill. 12-p catalog of pressure and vacuum gages for steam, water, air, oil, gas, ammonia, etc.

Heating: Commodore Heaters Corporation, 11 W. 42nd St., New York City. 16-p booklet on obtaining comfortable and economical heating with convector heaters, including installation suggestions.

Insulation: Geo. B. Smith Chemical Works, Inc., Springfield, Ill. 4-p pamphlet on insulating products—fill, cement, block,

(Continued on page 99)

Piping a Sand-Water Mix

By **C. L. Mantell***

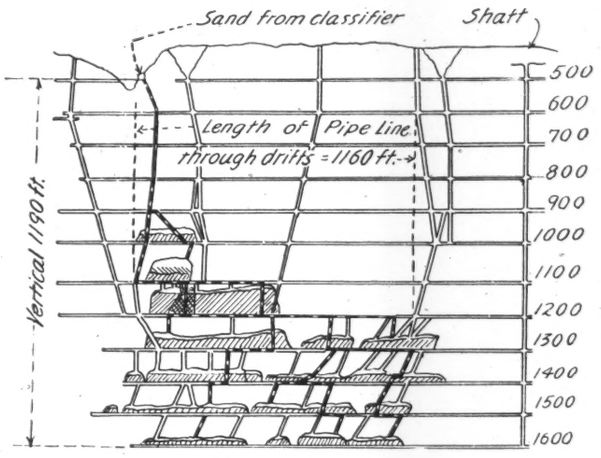


Fig. 1—Piping system for sand fillling of stopes in mine. The heavy line shows the piping

AT THE Pinar del Rio Matahambre copper mine in Cuba the ore is removed from the ore bodies underground and brought to the surface through a vertical shaft; this removal forms underground openings or rooms or *stopes*. The rock surrounding the ore is not strong enough to prevent cave-ins so the stopes are filled with the waste product or tailings produced on the surface as the result of working the ore. The material pumped into the mine for fill runs 30 to 65 per cent solids, the remainder being water. It has just enough fluidity to enable its being pumped.

In the development of the piping system, four types of lines were used. In the first, 4-in. iron pipe was employed and the sand run into the mine by gravity. Due to the slow travel of the sand-water mixture through the pipe, the lines plugged and had to be periodically cleaned out at various sections. In a second layout 2-in. iron pipe was employed. It functioned satisfactorily but lasted for only 1,000 tons or less, failure occurring at the bends. This pipe was later replaced by 2-in. extra-heavy pipe with plain ends, coupled. Even this heavy material, however, was unsatisfactory, as the abrasive action of the sand wore through the pipe.

In a third installation 2-in I. D. 4-ply rubber hose, having a rubber lining ¼-in. thick, was used. All curves in the lines were covered with split pipe sleeves to prevent kinking of the hose. Couplings in the line were rubber-lined nipples. The hose usually failed just behind these couplings, an effect thought due to the offset in the inner lining between the hose and the nipple. This type of line transmitted 25,000 to 30,000 tons of sand before failure.

Experimental runs were also made with sections of 3½-in. rubber-lined iron pipe. The pipe had a ¼-in. wall, a ¼-in. rubber lining, and an inside diameter of 2½ in. A test piece carried 80,000 tons of sand and upon examination, although rusted on the outside, showed no apparent wear on the rubber lining. In test lines, pipe sections 10 ft long, connected together by rubber-faced flange couplings, were used. Due to the rubber on the face of the couplings, the direction of the line could be changed about 20 deg without bending the pipe. This was done by tightening the flange bolts on one side more than on the other. The rubber-lined pipe could be bent cold about 90 deg.

In the ultimate development, the system of mine filling shown in Fig. 1 was employed.[1] The results have been reported by D. D. Homer.[2] Standard steel pipe 3-in. in diameter, 12-ft lengths, having a ¼-in. rubber lining vulcanized or welded to the inside of the pipe, is used with ordinary flanges. The rubber lining is brought out over the face of the flange, Fig. 2, and acts as a gasket. If the rubber lining is not brought out to the face of the flange, the sand will cut through any other types of gaskets or joints. The effective inside diameter of the pipe is 2½ in. The lines handle 35 tons of sand per hr and apparently have a much higher capacity. Over a period of about 16 months more than 200,000 tons of sand were put through the lines with almost negligible wear on the rubber. Some sections subjected to severe service at bends passed more than 200,000 tons of sand before small holes were worn through at the bends.

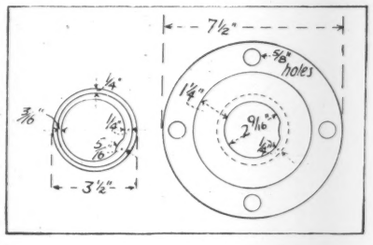


Fig. 2—Dimensions of rubber-lined pipe. Left, cross-section; right, flange end

The cost of handling the sand is low, being estimated at about 20 cents per ton, and the total cost for pipe estimated at 6 cents per ton of sand placed in the mine.

[1] Richert, Eng. Mining J., 127, 348 (1929).
[2] Eng. Mining J., 132, 367 (1931).

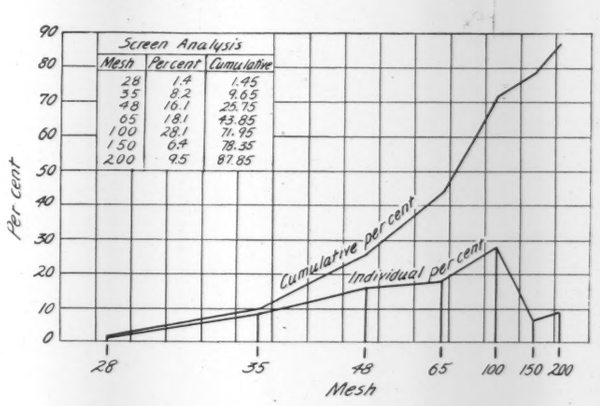


Fig. 3—Screen analysis of the sand pumped

* Consulting Engineer, Pratt Institute, Brooklyn, N. Y.

Designing Copper Gaskets for Hydraulic Piping

. . . In which a reader comments on a previously published article, and the author replies. Derivation of the Baumann formula used in design of flanged joints is also given.

WHEN designing copper gaskets by the method described in the article "Gaskets and Packing for Hydraulic Power"[1] the writer believes it would be preferable to compute the total load due to internal pressure and the total normal load on the gasket, rather than the load per inch of internal length. Dividing the sum of these two by the pressure required to make the gasket flow will give the area of the gasket, from which, knowing the internal diameter and area, the external diameter of the gasket can be obtained.

Several years experience with copper gaskets at high pressures since a similar method of design was described by Ernst, Reed, and Edwards[2] has indicated that this method is not correct for all sizes of copper gaskets. If calculations are made for various size gaskets it will be found that with small diameter gaskets the normal load is greater than the load due to internal pressure, while with large diameter gaskets the reverse is true. It is evident from this that the constant $C = 2.5$ can not be correct for all cases.

When figuring total bolt load, just why or how much the area of the gasket should be increased over the calculated area is not stated. The idea in assuming a larger gasket area is apparently to be sure of sufficient strength in the bolts, but this could be accomplished better by using a proper factor of safety.

The problem of designing gaskets and calculating bolt loads is not a simple one, especially for high pressures. The most difficult portion of this problem is to determine what load over and above the internal pressure load is required to make a tight joint. Different methods, ranging from a simple ratio to complicated formulas, are used for determining this so-called ratio of gasket load to internal pressure load—which of these is correct is a question. It is hoped that the results of tests now being made with copper gaskets will give some definite information.—*J. R. Dilley,* Mechanical Engineer, Department of Agriculture.

The Author's Reply

Mr. Dilley's comment is manifestly of a constructive nature, therefore it is most welcome. Let me preface this discussion by expressing regret that the work of Ernst, Reed, and Edwards was not properly acknowledged in my article.

The method of design proposed by Mr. Dilley might better have been illustrated by a sample computation in order to outline its desirable features more clearly.

In the Ernst-Reed-Edwards method the forces acting on the gasket are computed in accordance with the principles of mechanics, the equations being modified to correspond with experimentally established results. This procedure would appear to be sound, but Mr. Dilley indicates that unless values for the constant C vary with the diameter, the formula will be found incorrect. The writer believes that joints designed by this method should be satisfactory for all ordinary hydraulic work, and will concede that the method is inadequate for very high pressures or for very large sizes. For such conditions, however, *totally confined gaskets* should be specified.

Plastic Properties of Copper

Failure of a copper gasket, resulting from faulty design, may occur because the designer lacks the necessary data relating to copper which is squeezed beyond its elastic limit. The properties of copper below the elastic limit differ from those within the plastic range. Cold-working results in a structural change within the metal, *i. e.,* a reduction in grain size and hardening of the metal, followed by increased resistance to further deformation (due in large part to increase of area).

Table 1—Compression Tests of Sheet Copper[a]

Area of specimens, ¼ in. by 3 in. (¾ sq in.)

Load Lb per Sq In.	Original Thickness 0.103 In., Reduced to	Original Thickness 0.034 In., Reduced to
11,060	0.100 In.	0.03 In.
14,700	0.099	.....
18,000	0.098	.....
24,700	0.095	0.03
32,000	0.093	0.029
41,300	0.088	0.028
49,300	0.085	0.028
57,300	0.082	0.027

Area of specimens, ¾ in. by 3 in. (2¼ sq. in.)

Load Lb per Sq In.	Original Thickness 0.1025 In., Reduced to	Original Thickness 0.03 In., Reduced to
30,000	0.102 In.	0.030 In.
60,000	0.100	0.030
90,000	0.097	0.030

[a]The values given in this table are from compression test data obtained through the courtesy of the research laboratory of the Goetze Gasket and Packing Company, New Brunswick, N. J.

[1]Heating, Piping and Air Conditioning, March, 1932, p. 202.
[2]*Industrial & Engineering Chemistry, 17,* page 775, August, 1925.

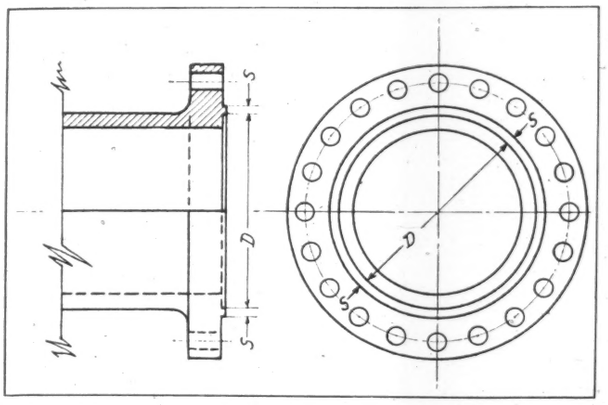


Fig. 1.—Diagram illustrating derivation of Baumann formula in text

Plasticity of copper also depends upon relative proportions, that is, a thin gasket will not yield to a pressure which causes a thicker one of equal area to flow. This is due to a pyramiding of pressure towards the center, as explained by E. V. Crane in his recent book "Plastic Working of Metals.[3]

Table 1 shows the greater strength of the thinner specimens, and the increased resistance to deformation with increase of area. The loads on the ¼ in. by 3 in. specimens were below the elastic limit, hence the compression is proportional to the load.

Design of Flanged Joints

In considering the problem of designing gaskets and calculating bolt loads, such items as gasket width, initial and maximum bolt stresses, flange stresses and flexibility, characteristics of materials, etc, require study and analysis. The more important variables associated with the design are:—cylinder and gasket areas, unit pressure of fluid and unit pressure on joint surface.

A comparatively simple formula expressing the relations between these variables was presented by Karl Baumann in his masterly address[4] before the *Institution of Mechanical Engineers.* Although developed for use in designing steam-pipe flanges, the formula has a more general application. By its use, the total bolt load for any given set of conditions may be calculated in terms of the thrust upon the inside area of the pipe or cylinder. One assumption is necessary in applying this method, and that is made in determining the optimum ratio between unit pressure on joint surface and unit working pressure. (For steam engineering, Baumann specifies a minimum ratio of 2:1 as the requirement for maintaining a tight joint.)

Referring to Fig. 1, the following steps show the derivation of the formula.

Let P=Working pressure in lb per sq in.
L=Total bolt load in lb.
=Pressure over joint surface.

Let $m=\dfrac{\text{Unit pressure on joint surface}}{\text{Unit working pressure}}$

$L=K\times\frac{\pi}{4}D^2P$

The annular area of the joint surface (gasket) is:

$$\frac{\pi(D+2S)^2}{4}-\frac{\pi D^2}{4}=\frac{\pi D^2}{4}+\frac{4\pi DS+4\pi S^2}{4}-\frac{\pi D^2}{4}=\pi S(D+S)$$

$$L=\frac{P(D+2S)^2\pi}{4}+Pm[\pi S(D+S)]=P\left(\frac{D^2\pi}{4}\right)\times K \quad\ldots\ldots\ldots\ldots 1$$

$$=(D^2+4DS+4S^2)\frac{\pi}{4}+m\pi DS+m\pi S^2=\left(\frac{D^2\pi}{4}\right)K \quad\ldots\ldots\ldots\ldots 2$$

$$K=\frac{D^2+4DS+4S^2+4mDS+4mS^2}{D^2} \quad\ldots\ldots\ldots\ldots 3$$

$$K=1+\frac{4(mDS+DS+mS^2+S^2)}{D^2} \quad\ldots\ldots\ldots\ldots 4$$

$$K=1+\frac{4[SD(m+1)+S^2(m+1)]}{D^2}=1+4(m+1)\frac{SD}{D^2}+\frac{S^2}{D^2} \quad\ldots\ldots 5$$

Baumann's Formula: $K=1+4(m+1)\dfrac{S}{D}\left[1+\dfrac{S}{D}\right]\ldots\ldots 6$

In Equation 1, the expression $\dfrac{\pi P(D+2S)^2}{4}$ represents the thrust due to liquid pressure exerted upon the total area within the outer circumference of the gasket, and $Pm[\pi S(D+S)]$ is m times the pressure exerted upon the gasket area.

If, in Fig. 1, we assume values of ⅞ in. for S, and 19¼ in. for D, then $\dfrac{S}{D}=0.0455$.

Then, when $m=0$, $K=1.19$
$m=2$, $K=1.57$
$m=4$, $K=1.95$
$m=12$, $K=3.47$
$m=16$, $K=4.23$

It will be noted that the formula includes the gasket width S, and that only simple computations are required. Excellent judgment, based upon experience or investigation, is necessary, however, in making the assumptions on which important designs are to be based.

Unusual Designs Cited

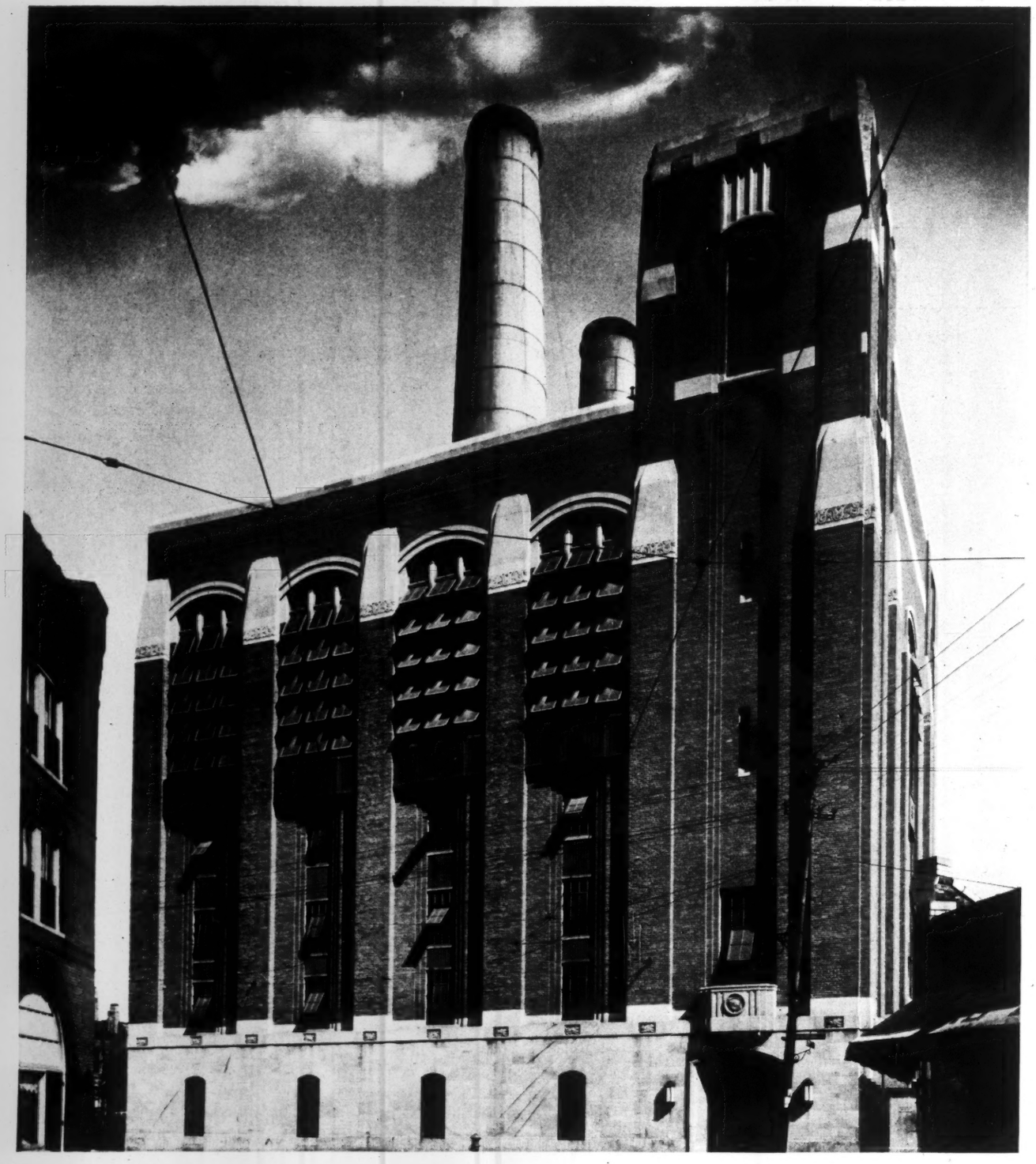
Although hydraulic pressures are very much greater than those employed in steam engineering, joint design for the latter is much more difficult because of accompanying high temperatures. Some recent designs for high pressures and large sizes for both steam and hydraulic apparatus have been notably successful. A steam-to-steam reheater operated at 1400 lb. W.S.P. and 750 F, with joints about 5 ft in diameter, is giving excellent service. Hydraulic apparatus, invented by F. C. Langenberg, has cupped packings about 2 ft in diameter, and withstands pressures as high as 50,000 lb per sq in. This apparatus is not for laboratory purposes, but is used for cold working large metal molds.—*F. E. Wertheim,* Mechanical Engineer, Chicago, Ill.

Text on Use of Coal

In "Modern Combustion, Coal Economics, and Fuel Fallacies" Clarence V. Beck discusses the mining of coal and the mining industry, compares coal with other fuels in use today, and, in Part III, devotes 28 chapters to combustion. The section on combustion includes information on the fundamentals, boiler efficiency, power as a byproduct of process or heating steam, stokers, heating ills, combustion instruments and their uses, etc. An appendix explains steam power efficiency and the steam table, and contains tables of comparative fuel values and degree-days.

The publisher is Mid-West Coal Retailer, Chicago, Ill.

[3]John Wiley and Sons, 1932.
[4]Proceedings of the *Institution of Mechanical Engineers,* December, 1930, p. 1317.



The beautiful new central heating plant for the University of Chicago which replaces the plant established in 1894. The plant is adjacent to the Illinois Central to cut coal delivery cost

Central Steam Plant and Pipe Tunnels Keep Heating Costs Down at U. of C.

The new central heating plant and tunnel piping system at the University of Chicago have been in operation 3 heating seasons and are constantly being developed. The recently-opened International House is the latest building to be heated from this plant.

The new plant and piping were constructed because:

Heating costs were mounting because of obsolescence of the old plant;

The coal bill was excessive due to haulage costs;

The building program required new steam lines.

Among the problems solved in the design of the system was the method for returning condensation from the various building heating systems — some of them one-pipe gravity and the newer ones two-pipe vacuum —to the central plant. This and the other features are described here by Charles Levin*.

ABOUT 1894, the University of Chicago established a central heating plant for its buildings. Steam was distributed through piping carried in a series of tre es and underground duct lines and, with some minor changes in the runs, served until the new plant was built. Up to 1928, the University's new buildings were constructed relatively close to the existing trenches, and new sections to serve them were added as required. However, the University found, as time went on, that:

The heating plant costs were increasing tremendously due to the abnormal amount of repairs on the old equipment.

The coal bill was increasing out of proportion to the amount of steam used, due to the cost of hauling.

The building program brought them south of the Midway where there were no steam lines.

After due consideration to the problems involved, it was decided to build a new central heating plant in a more advantageous location which would release the ground occupied by the old boiler plant for new educational buildings; a site along the Illinois Central tracks at Blackstone Ave. and 61st St. was selected.

Tunnel Piping System Chosen

Two possibilities of steam distribution to the campus presented themselves: either a series of cast iron and tile ducts, or a main tunnel large enough to accommodate all the piping, with working space sufficient for the maintenance requirements. By investigation, it was found that the initial cost of a main tunnel would be greater, but, as the load increased, more ducts would have to be added, the cost of installation and maintenance of duct lines over a period of time would exceed that of the main tunnel, which would require only additional pipe lines to care for increased demands. The following table shows the load requirements:

Load Requirements

Heating Season	Total Radiation in Square Feet	Building Contents in Cubic Feet
1920-21	364,552	20,987,897
21-22	365,452	21,015,947
22-23	370,222	21,503,947
23-24	372,236	21,721,587
24-25	367,749	21,404,602
25-26	404,280	23,404,681
26-27	486,527	28,353,281
27-28	550,000	31,995,921
28-29	610,000	35,428,605
29-30	690,000	40,062,205
30-31	802,000	46,503,305
31-32	850,000	49,300,000
32-33	885,000*	51,300,000
33-34	920,000*	53,300,000
34-35	955,000*	55,300,000
35-36	987,000*	57,300,000
36-37	1,025,000*	59,300,000
43-44	1,110,000*	64,300,000

*Estimated.

The construction of the tunnel is monolithic reinforced concrete modified in places to meet varying conditions. Considerable difficulty was encountered in the portion under Blackstone Ave. where the street railway track had to be removed, adjacent buildings supported by shoring and sheet piling driven to prevent slides. The section at this point is heavier than normal owing to the ever-increasing loads from street cars and trucks. Under the bridle path, Midway and the campus, the normal section is used, except where the entire tunnel is under water. Under street crossings, the section is slightly heavier to take care of increased loadings.

Entire Tunnel Waterproofed

The tunnel was built in sections approximately 50 ft long. To provide for expansion, a ¾-in. caulked joint was made. Due to interferences from sewer, water, and gas lines at street intersections, the tunnel in some cases had to be constructed below water level. It was decided to waterproof the entire tunnel. On the top surface of the first section of the tunnel, and two feet down the sides, there are one to three layers of waterproof membrane, mopped on and coated; plastic waterproofing was troweled on the interior walls. The second portion of the tunnel has an integral waterproofing. At various points throughout the entire length of the tunnel, expansion areas, manholes, vent openings, and removable slabs for easy access in removing and replacing pipes are provided. Vent openings are located at each expansion area. While at present gravity ventilation is used, the openings are arranged for the installation of high-speed disc fans.

To take care of the drainage, the floor of the tunnel is pitched to a central gutter and the floor pitched to sumps where automatic electric bilge pumps have been installed which discharge into the nearest city sewer line.

Pipe Sized for Future Load

Estimates of the load requirements to 1944 were made, as shown in the preceding table. In design, a single line of pipe was provided of capacity to care for some 55,000,000 cu ft of building space. It was found that the saving in installing piping at this time for a

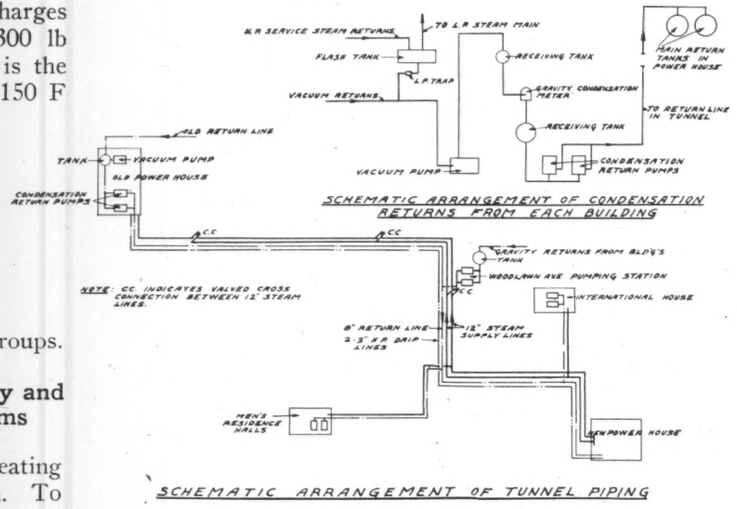
*Neiler, Rich & Co., Consulting Engineers, Chicago, Ill.

greater load would be less than the carrying charges on the investment. Piping was designed for 300 lb pressure and 150 degrees superheat, although it is the intention at some future date to use 175 lb and 150 F superheat. At present, 150 lb and 150 F is being used.

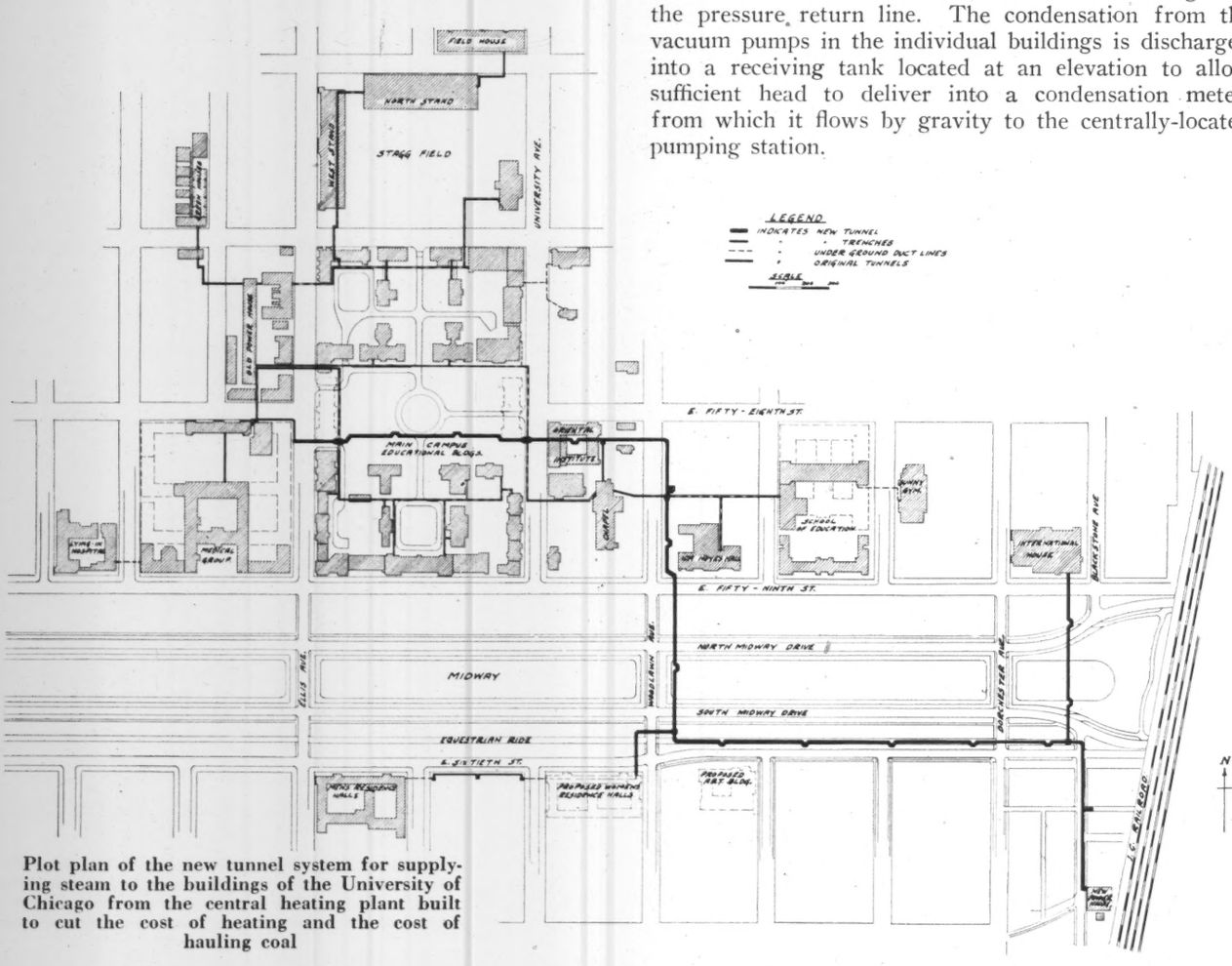
Duplicate 12-in. steam lines run from the header in the central plant through the entire length of the main tunnel. At certain points cross connections are made between the two lines to permit repairs without shutdown of the entire system. Sub-feeds are taken off at various points to furnish steam to certain building groups.

Returning Condensation from One-Pipe Gravity and Two-Pipe Vacuum Building Heating Systems

The older buildings have one-pipe gravity heating systems, while the later ones are two-pipe vacuum. To return condensation of all building systems to the central plant, the following arrangement is used: The piping in the original north campus area has been rehabilitated to collect the returns from the various buildings and empty the condensation into a receiving tank located in the old boiler-house (now the Service Building). A pump maintains a vacuum in this tank and the water is removed by condensation pumps which deliver the condensate into the pressure return line to the new central plant. The other pumping stations are provided with receiving tanks taking the condensation returned for each building. These returns from the new buildings are discharged by gravity to the various centrally-located condensation receivers, and thence discharged to the pressure return line. The condensation from the vacuum pumps in the individual buildings is discharged into a receiving tank located at an elevation to allow sufficient head to deliver into a condensation meter, from which it flows by gravity to the centrally-located pumping station.


SCHEMATIC ARRANGEMENT OF CONDENSATION RETURNS FROM EACH BUILDING
SCHEMATIC ARRANGEMENT OF TUNNEL PIPING
NOTE: CC INDICATES VALVED CROSS CONNECTION BETWEEN 12" STEAM LINES.
WOODLAWN AVE PUMPING STATION
INTERNATIONAL HOUSE
NEW POWER HOUSE

The upper diagram shows how condensation from each building is returned to the central plant. The lower diagram shows how the returns from the older buildings are collected at the old power house—now the service building—for return to the new plant



Plot plan of the new tunnel system for supplying steam to the buildings of the University of Chicago from the central heating plant built to cut the cost of heating and the cost of hauling coal

Removing Condensation from Lines

For removing condensation from the transmission lines (which will occur when first starting up the system with the piping cold) return traps are provided; these discharge into the main 3-in. return pipe. After the system is once in service, there is practically no condensation—due to the superheat—but should there be a slight amount which would naturally collect at points where the pipes are low, these same traps take care of it.

Piping Materials and Insulation

The 12-in. supply pipes are extra heavy wrought steel and the 8-in. return is extra heavy genuine wrought iron; the drip lines are full weight genuine wrought iron. All joints are welded. At the expansion areas, flanged pull-up joints are provided. Where fittings are installed, extra heavy cast steel with Sarlun flanges were used.

The high-pressure steam lines are covered with a 1½-in. layer of high-temperature pipe covering, securely wired in place, followed by a 2-in. layer of 85 per cent magnesia. All joints were broken and sealed with asbestos cement before the sections were drawn tightly together. All the valves and fittings are covered with 1½-in. high temperature blocks securely wired and built up with asbestos to the thickness of adjacent covering. The drip lines are covered with two standard thick layers of 85 per cent magnesia and the 8-in. return line is covered with a standard thickness of 85 per cent magnesia. All the covering is jacketed with 8-oz canvas.

Expansion of Piping Measured

U-bends care for expansion of the piping. An indicator is welded on each line to measure the expansion in that section of piping. It is interesting that when the expansion of the 12-in. high-pressure line for a run of 1700 ft was checked by these indicators, the amount of expansion was found to be 3.41 in. per 100 ft as compared with the calculated expansion of 3.38 in. from the formula:

$E = C \times (T - t) \times L$.

E=Expansion in inches.

$T - t$=Temperature difference.

L=Length of pipe in inches.

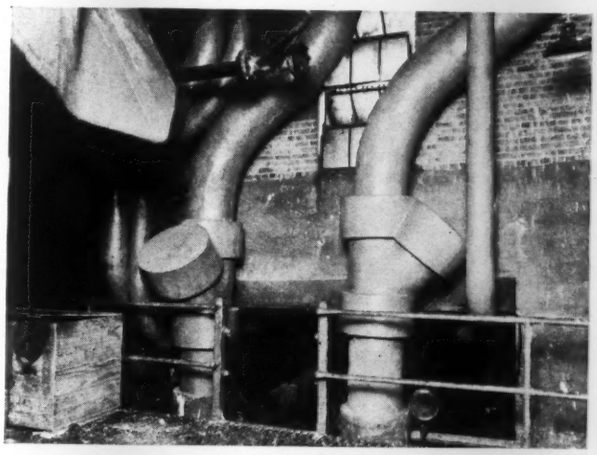
C=The coefficient of expansion of the metal of which the pipe is made.

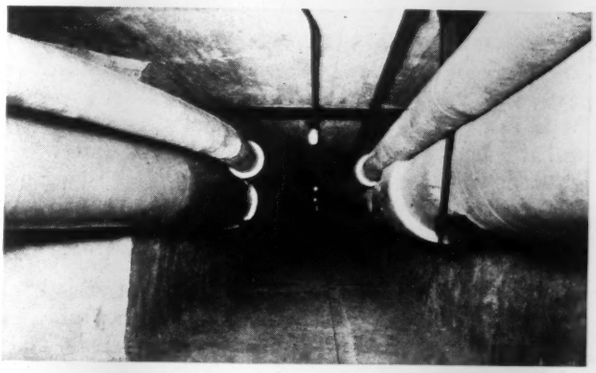
The tunnel is wired throughout for electric light and power. A telephone system is provided between the service building and the new central plant, with intermediate stations located at various points in the tunnel.

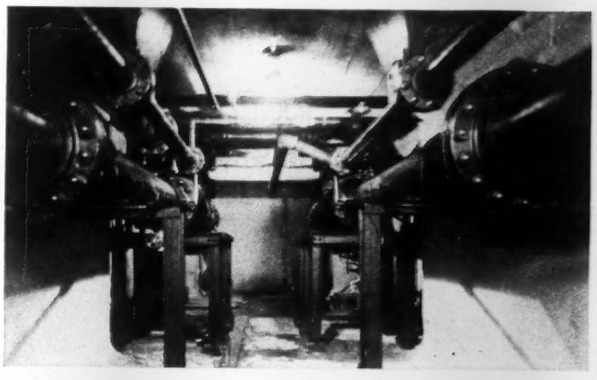
While the tunnel has been in service only a comparatively short time, the results show that the method of construction and the system of steam distribution is successful and eminently satisfactory.

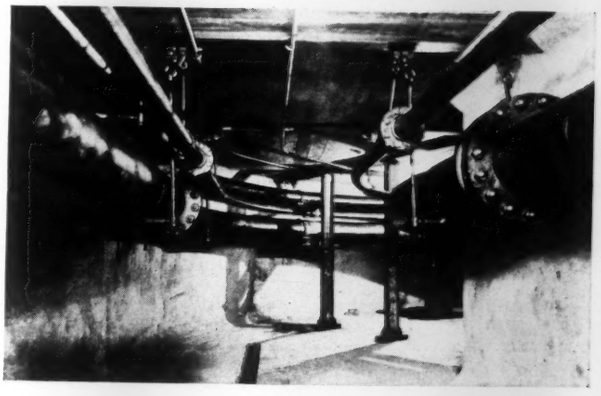
Acknowledgement is made to S. G. Neiler, and E. P. Rich, of Neiler Rich & Co., the consulting engineers, and L. R. Flook, general superintendent of buildings and grounds of the University of Chicago, for their co-operation in making available the data used in the preparation of this article.

Top to Bottom—The steam lines entering the tunnel . . . a straight run . . . Anchor points in the tunnel . . . The tunnel piping just before an expansion bend









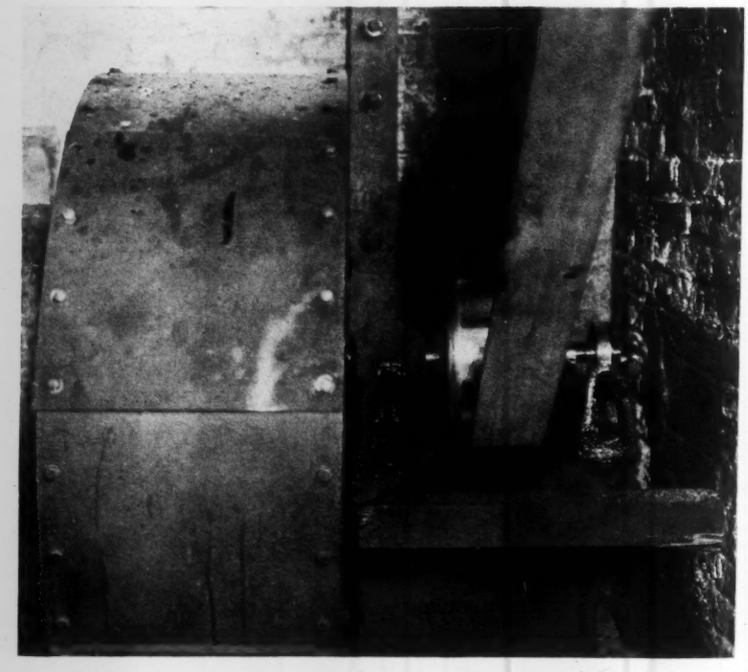
Foundry Saves $6000 Annually in Exhaust-Fan Bearing Maintenance

BY modernizing with dust-tight heavy-duty self-aligning adapter-type ball bearings on 13 exhaust fans the Ferro Machine and Foundry Company of Cleveland has realized a saving of about $6,000.00 a year. At this rate the new bearings pay for themselves every two months. This is modernization with a vengeance and of a kind which no industrial management can afford to overlook.

The service conditions of the exhaust fans are extremely severe, as they remove dust and grit from rotary mills and sandblasts for cleaning castings in the foundry. They are all located in a central blower house where the dust is collected in hoppers and discharged by gravity into cars. Normally the fans operate 24 hours a day in a dust- and grit-laden atmosphere. It is extremely important that they operate continuously and without interruption and as a result constant attendance is required. With the old bearings the oil consumption was from five to six gallons per day and two men per shift were kept on duty. Bearing metal had to be renewed every two weeks on the large fans and every two months on the smaller ones; usually on Sundays because of the need of keeping the fans in continuous operation.

Realizing the excessive expense of this, ball bearings were installed on two of the fans for trial. During this five-month period careful records of performance and cost were kept. The results were so satisfactory that at the end of that period all of the fans were similarly equipped, a simple job.

The dust-proof housings have kept out the grit and dust which formerly ruined the old bearings and ten years has been assumed as a conservative life for the new bearings. The consumption of lubricant was reduced from 5 gallons of oil per day, costing some $250.00 per year, to 25 lb of grease per week, costing $97.50 a year. One man per shift cares for them. The rebabbitting charge of over $2,300.00 per year has been eliminated. A careful analysis making allowances for depreciation, maintenance and other annual charges has shown a net saving due to the new bearings of almost $6,000.00 a year. This is without crediting any allowance for reduced power consumption or reduced wear and tear on the belt drives.—*Francis A. Westbrook.*



This view illustrates the conditions under which the average exhaust fan must operate. The housing between the pulley and the fan shows the pillow block covered with oil and muck; such conditions impose severe requirements on bearings. At the right, the bearing cap was removed to expose the bearing

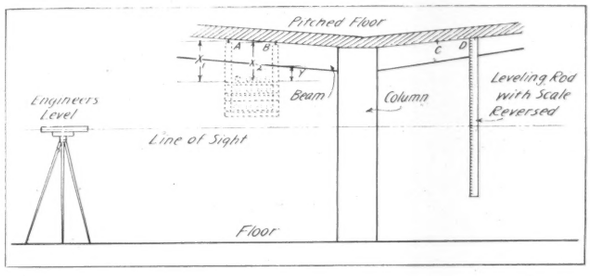
ON THE JOB

. . . a department devoted to short, practical articles on the installation, operation and maintenance of air conditioning, heating and piping equipment . . . . . .

Hanging Thirty Miles of Brine Piping

ON a recent job we had thirty miles of brine piping made up into ceiling coils to erect and of course wanted to keep the costs down and still make a good job. Matters were made more difficult by the fact that the ceilings, which were of concrete, pitched very sharply and many cast beams were used in the construction.

We could have used threaded rods with plenty of threads on them and pipe sleeve separators betwen the hanger bars, or we could fabricate hangers of structural steel as shown in the sketch and bend over the upright members on the job after first running a line through the rooms to determine the dimensions X and Y. Either of these methods would have been satisfactory but as there were a great many hangers involved a new method of erection was utilized.



Showing use of level in figuring hangers for brine piping ceiling coils

It was decided to use an engineer's level and a leveling rod made to order whereon the numbers ran from top to bottom instead of from bottom to top as is usual. The inserts were numbered on the blue prints and the insert levels shot, as at inserts A, B, C, and D as shown in the sketch. The lowest beam points were established—also bench marks—so that the level could be moved around. Dimensions X_1 and X_2 could then be calculated and the hangers completed at the home plant, with special shop equipment, at small cost. The hangers were then painted and numbered to correspond to the numbers on the drawing and shipped to the job. The work was a complete success.—*Robert S. Wheaton.*

Modernizes Fan and Compressor Drives

AN early step in the modernization program of an Indianapolis automobile body plant, started about a year ago, was the bringing together in a central location of two compressors; previously the compressors had been located in out-of-the-way spots at considerable distance from each other in the plant. One was driven by a conventional fixed-center drive, the motor being 30 ft from the compressor, and the other had the usual idler drive.

Re-location of the two compressors in a well-arranged and lighted compressor room was made possible by the application of the flat-belt short-center drives with which both were equipped. As shown in Fig. 1 the motor is mounted on a free-swinging pivot shaft with its weight cradled in the belt to maintain the proper belt tension. Adjustable arms permit the motor to be shifted the correct distance from the pivot shaft to maintain the right belt tension for the particular load. (When a belt comes up to speed, centrifugal force lengthens it, necessitating some means of taking up the slack.)

By this close grouping of these two compressors and their drive equipment, maintenance expense has been largely eliminated and the savings in belting are expected to pay for the new equipment in the first year.

New Fan Drive Saves Power

Just recently the woodworking department of this same plant has been brought up-to-date and modernized. The fan for the dust and shavings collection system has also been equipped with a short-center, flat belt drive as shown in Fig. 2. This fan was formerly driven by a vertical belt with a screw-adjusted idler operating against the slack side. The belt descended into the basement to

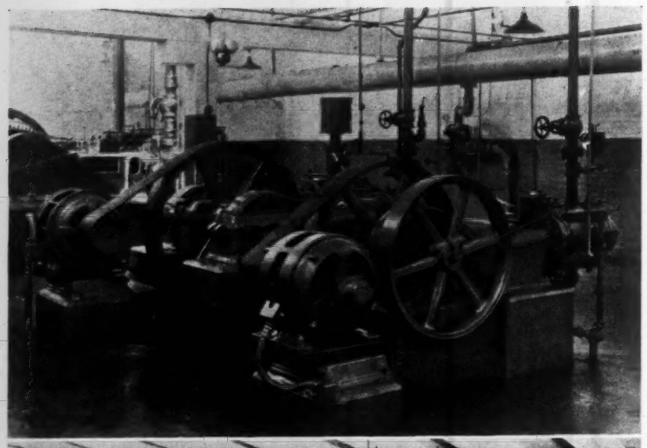


Fig. 1—Two compressors, formerly located in out-of-the way spots in large plant, now in well-arranged compressor room and equipped with flat-belt short-center drives

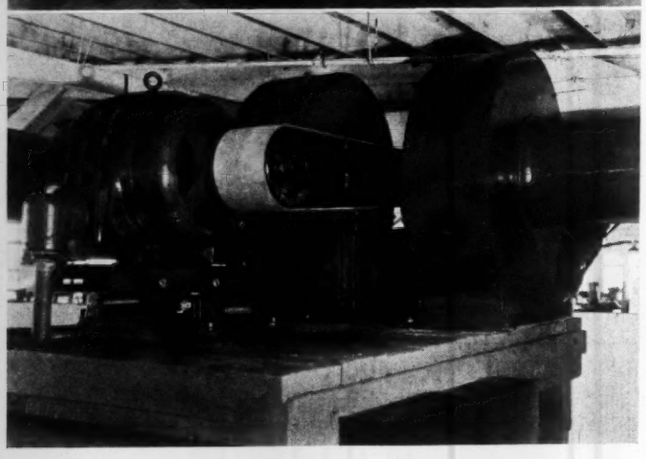


Fig. 2—Fan for dust and shavings collection system in woodworking department. The short-center drive eliminated 100 ft of belting, 20 per cent slip, and a maintenance problem*

a jack shaft located on the floor, the jack shaft in turn being driven by a long, flat belt. Pulleys on the fan, jack shaft and motor were metal and wood. Almost continual maintenance attention was required; the new drive has eliminated 100 ft of belting and solved this maintenance problem.

Another advantage (demonstrating the power that can be wasted by inefficient operation) was obtained: After the new drive was installed it was found that the fan speed was 20 per cent higher than was required, although pulley sizes were such as to give exactly the same speed as should have been had by the old installation. In other words, 20 per cent slip—which means wasted power—had been eliminated by the drive engineered for the work to be done.

Experts Discuss Valve Installation

HOW best to install valves to meet various operating conditions is a subject of prime interest to piping maintenance engineers; the following comments on this matter are the result of an article by W. H. Wilson on installing globe and angle valves which appeared in the June, 1932, issue. They represent the views of five experts.

Comment No. 1

Referring to the article on installing globe and angle valves which was published in the June, 1932, HEATING, PIPING AND AIR CONDITIONING, our engineers have the following comments:

Fig. 1*A* (see next page) shows the most practicable and satisfactory method of installing angle valves for the reason outlined in this article. Fig. 1 shows an installation of an angle-type valve for boiler blow-off service. For this service it would probably be better to use a valve of the quick-opening type, giving full pipe opening with no restrictions. It is also our opinion that the quick-opening valve should be backed up with a gate valve or stop cock of a lubricated type. This would eliminate any wire drawing of the blow-off valve due to impurities in the boiler water.

If the valve were installed opposite to the way it is shown in Fig. 2, we do not believe the valve would seat itself should the disc become detached. It would, however, seat itself if the valve were installed in a horizontal position.

The installation as shown is correct, but we would recommend the use of a gate valve for this type of service, because the feed line is usually throttled at the point where water enters the boiler drum, and not at the pump. A gate valve would give full pipe opening at discharge and would also eliminate the resistance of flow through the globe type valve.

The installation at *D* in Fig. 3 is the most practicable method to use on a feed line. Installation *C* would endanger the boiler should the valve disc become detached. The fire would have to be pulled because of lack of water and at the same time the boiler would have to be taken out of service.

Installation *H* in Fig. 4 is the most practicable method, but for exhaust steam we would use a gate valve installed upside down, even though the valve bonnet would form a pocket. This would do no harm, except to ac-

*Illustrations from F. W. Parkhurst, The Rockwood Manufacturing Co., Indianapolis, Ind.

cumulate a little lubricating oil which comes over with the exhaust steam. The reason for installing the gate valve upside down is to eliminate dropping of the gate. This is likely to happen because exhaust lines are subject to pulsations from pumps, compressors, etc., which would keep gate vibrating. The gate valve would also reduce resistance.—*J. B. Gibson,* Western Electric Company, Chicago, Ill.

Comment No. 2

We offer these comments on the installation of globe and angle valves as they are written up in the article by Mr. Wilson in your June, 1932, issue.

We believe that for most cases the principle to be followed in installing a globe or angle valve is pressure under the disc. An important reason for this is that when pressure is under the disc and the valve is closed, the pressure then is not on stuffing box and packing which is an important point, particularly in high-pressure installations. Another point in favor of pressure under the disc is that when the globe or angle valve is used for throttling there is a closer control and less likelihood of rattling or chattering.

In justifiable defense of valves generally, I would say that the danger of the disc coming off the valve is remote. This reason for installing with pressure below the disc is one that goes back to an earlier day before manufacture had reached its present state of dependability and before designs had developed to their present merit.

As a condition creating an exception to this general principle of installing with pressure under the disc with very high pressures, it is sometimes advisable to install with pressure above the disc in order to keep the valve tight. Some manufacturers have made such recommendations in their printed specifications.

As a point away from the main issue of the article, I would comment on the illustration shown of the blowoff hookup in Fig. 1. This shows, following the piping from the boiler—first, a regulating or throttling element (angle valve) and, then, a shutoff element (usually a cock). It is our opinion that the shutoff element should be between the regulating element and the boiler. The reason for this is that the regulating element takes most wear and more often needs to be repaired or replaced.

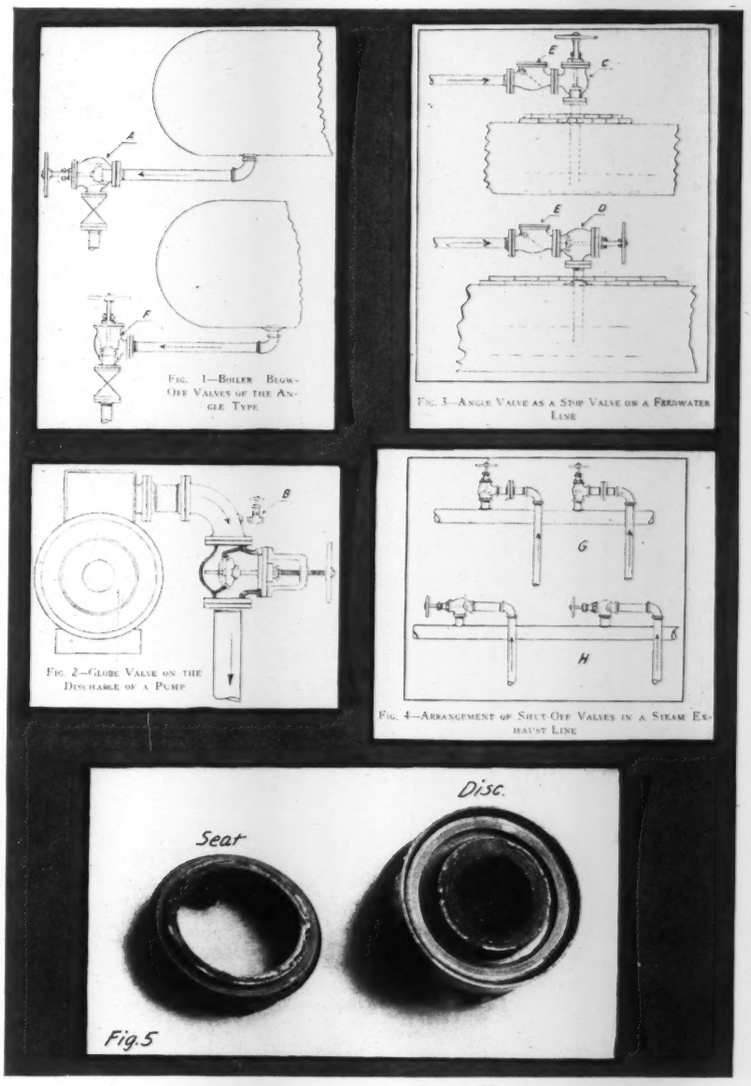
If the shutoff element is next to the boiler, this repair or replacement of the regulating element can be taken care of without throwing the whole equipment out of service, but if the regulating element is next to the boiler, it can only be repaired or replaced when the boiler is out of service. —*J. P. Ferguson,* Reading-Pratt & Cady Company, Inc., Bridgeport, Conn.

Comment No. 3

In general, it is quite correct to say that the pressure side is the under side of the disc, that is, the side opposite to the stem. In other words, the flow through the valve should oppose the closing of the valve.

However, there are exceptions to this rule. For example, if the valve is regularly used for throttling, with a large pressure drop at the valve seat, as in a continuous blow-down to atmosphere, the valve should be reversed. Otherwise the high velocity conical jet between the seat and disc, bearing particles of water and other material, impinging on the interior of the valve body above the seat will cause rapid erosion of the body metal, resulting in failure of the valve. With the valve reversed this conical jet is converging below the seat so that the erosive particles are diverted from their path and the wear of the valve is reduced to a minimum.

In any event, the secure attachment of the disc to the stem is a matter of fundamental importance in valve construction.—*F. B. Corey,* Pittsburgh Valve & Fittings Company, Barberton, Ohio.



Comment No. 4

Mr. Wilson's suggestions concerning the installation of globe and angle valves are backed up by sound reasoning and well propounded conclusions. In the first example, however, he states that a drain valve should be installed with the pressure under the disc due to the fact that the valve may be easily repacked when the disc is seated. The opposite view in this case is given as follows:

A. Valves on drip lines, valves on blower, soot blower or blowoff service, and shut-off valves for steam and other high-temperature conditions should be installed with the pressure on top of the disc. Under any of these conditions there is a cooling down due to collected condensation when the valve is closed after being operated and this causes the stem to contract. With the pressure on top of the disc, the tendency will be to hold the valve tight, permitted by the lost motion in the valve parts. However, if the pressure is underneath the disc, the stem will cool to a lower temperature than under the first condition and probably cause leakage at the seat. In the latter case also, the pressure being under the disc tends to hold the disc away from its seat after the stem has contracted and permits leakage, resulting in wire drawing.

The principal argument generally advanced against installing a valve with pressure on top of the disc is that the stuffing box will always be subject to pressure and cannot be repacked whether the valve is opened or closed. The fact still remains that when a stuffing box leaks, it can often be tightened by pulling down on the nuts and if it cannot be entirely stopped in this manner, it is an indication to the operator of trouble and he will correct the condition the first time the installation is shut down. On the other hand, if the valve seat leaks due to the fact that it was installed with pressure under the disc, it will wire-draw and the operator probably know nothing about it until it gets so bad that it becomes a detriment. When this condition exists, nothing can be done to correct it until the installation is shut down and then an expensive repair is necessary and may even require a new valve.

The following additional suggestions for installing globe and angle valves cover a great majority of cases:

B. When a valve is used to throttle, the pressure should be on top of the disc. Under this service condition the valve is subject to vibrations and steam pulsation which may tend to loosen the parts or cause wear and failure in time. If the disc connection fails or the threads strip, the pressure would force the disc to its seat and close the valve. If installed with the pressure under the disc, the same trouble might permit the disc to tilt and wedge so that the valve could not be closed.

C. Valves used in boiler-feed lines and on the suction or discharge side of any oil or water pump should be installed with the pressure under the disc. With the pressure on top of the disc, and the valve partly open, every stroke of the pump will cause a water hammer blow against the upper side of the disc, tending to pull it away from the stem, thus endangering the stem and disc connection. If failure should occur, the water supply would be cut off and probably cause a boiler failure.

D. When a globe valve is used between a boiler and the main header, the boiler pressure should be under the disc. When thus installed, if the boiler is cut out, the valves will tend to remain tight due to the header pressure on top of the disc. The valve can be repacked by backseating the disc while the boiler is "in". This installation is, however, generally avoided, a gate valve being used in preference to a globe valve.

E. As a general rule, a globe valve can be closed more easily with the pressure on top of the disc than under the disc, but it is more difficult to open a valve with the pressure on top of the disc, especially in large sizes, unless a bypass is used to equalize the pressure. A valve 6 in. or larger for pressure over 125 lb should never be installed with the pressure on top of the disc, unless it is a balanced valve or the pressure can be equalized on both sides of the valve before operating.

Conclusion: Generally speaking, all drip, drain and blowoff valves should be installed with the pressure on top of the disc.

In an installation where it would be desirable to have the flow stop in case of a stem and disc connection failure, as in the throttle valve, the pressure should be on top of the disc. When it is desirable to have the flow continue in case of damage to this connection, the valve should be installed with the pressure under the disc.—*A. M. Houser,* Engineer of Product, Crane Co., Chicago, Ill.

No. 5—The Author Comments

Mr. Gibson states in regard to Fig. 1 that a quick-opening valve is suitable for this installation. The use of a quick-opening valve in this case is in line with many of the present day boiler installations. Valves of this type are made by many of the leading valve manufacturers.

Regarding Fig. 2, it is stated that the installation of a gate valve is recommended. We have installations similar to this, with a gate valve installed, as suggested. The seating faces of the gate valve are worn badly, making it difficult to work on the boiler feed water pump, on account of the leakage of hot water. We have a globe valve ready to put in place of the gate valve. We have experienced this trouble with gate valves on similar installations. The gate valve has the advantage of a straight, unobstructed passage for the flow. The flow through the globe valve is obstructed and strikes the disc. A gate valve is not easily reseated when in position on a boiler-feed line. For this reason we give the globe valve the preference on this class of work. It is a matter of opinion, we think, as to whether a globe valve or a gate valve should be installed at this location.

Fig. 3. Installation on boiler feed water line. We agree with the writer.

Fig. 4. Installation H: A gate valve could be used to advantage at this location. Low-pressure exhaust steam does not cut the seating faces of a valve as much or as quickly as boiler feed water. We would not place the gate valve upside down. The pocket formed by the bonnet of the valve would collect solid matter and impurities that would interfere with the operation of the stem and gate. The lubricating oil coming through with the exhaust steam, would, in our opinion (the pocket in the bonnet being full of water), float on the water and pass out throngh the pipe line. There is a point in favor of the installation of the valve upside down, when the gate is detached and drops out of the way. On the other hand when a gate valve is installed with the stem upward when the valve is open, the gate is drawn up and out of the way of the flow through the valve. The gate, in this position, is not subject to the vibration or force of the flow. There is little chance of the gate becoming detached and causing trouble.

Regarding the comment from Mr. Ferguson, manufacturers have improved and strengthened valves including more securely attaching the disc to the stem, although

discs occasionally become detached. This is due sometimes to the parts being worn away and weakened by corrosion and at other times by pounding and chattering, due to the flow and certain other operating conditions.

As to the last paragraph of this comment, regarding the tandem hookup, one of these valves is not shown in detail. This is a valve of the globe pattern. If a shutoff element such as a stop cock is used at this location it could be placed on the boiler side (nearest to the boiler).

Regarding the comment from Mr. Corey, he brings out a point in favor of installing the globe valve with the pressure or flow on top of the valve disc, when the valve is used for throttling. He also mentions an important matter when he states: "In any event the secure attachment of the disc to the stem is a matter of fundamental importance in valve construction."

A Boiler Blow-Off Valve:

Amplifying my original article regarding the installation of valves of the globe and angle pattern on boiler blowoff lines, it has been suggested by some operating men that the stream of water containing sediment and scale flowing with high velocity through the blowoff pipe and striking the bottom of the valve disc will destroy the seating faces of the disc and cause the valve to leak. Manufacturers of valves have designed seats and discs that are protected from this condition. Fig. 5 verifies this. The unretouched photo shows a valve seat and disc of monel metal that was in use for some time in a boiler blowoff valve. In the picture the valve seat is shown at the left and the disc at the right.

The small white spot on the seating face of the disc, at the top and to the right of the center line of the picture of the disc, is the only imperfection that caused the valve to leak slightly. This was easily taken care of by "grinding in" the seat and disc with valve grinding compound. The center or plug portion of the disc is worn badly and cut by the force of the water and scale striking the disc with high velocity when the valve was opened for blowing down purposes. This plug portion of the disc, in connection with the raised ring portion of the seat, protects the seating faces. This picture shows a good example of the improvements in the protection of the seating faces of valve seats and discs that are used where the conditions are severe.

Drip and Drain Valves:

Mr. Houser, explains his views clearly and covers the subject well. He recommends that drip and drain valves be installed with the pressure on top of the disc. Lately we have commenced installing globe valves in this manner and have not had sufficient time to compare results with former installations. Among the places where this method is followed are, on the blow down pipes from boiler water columns and drop pipes from overhead steam lines, used for test connections. In both of these cases the drop pipe fills with water of condensation when the valve is closed.

I am pleased to have the privilege of reading these comments. The points brought out are interesting.—*W. H. Wilson,* Member of Board of Consulting and Contributing Editors.

Recent Trade Literature

(Continued from page 86)

board, brick, mortar, pipe covering—of aluminum magnesium silicate with fusion temperature of 2485 F.

Motors: General Electric Company, Schenectady, N. Y. 52-p book on sychronous motors including economics of synchronous motors, description, characteristics of high- and low-speed motors, special motors, power factors, control equipment, applications to various industries.

Motors: The Lincoln Electric Co., Cleveland, O. 4-p pamphlet describing features of motors of arc-welded construction.

Process Equipment: The Patterson Foundry & Machine Company, East Liverpool, O. 8-p well-illustrated bulletin on special industrial process equipment such as mixers, jacketed kettles, agitators, special pipe and fittings, etc.

Pumps: Roots-Connersville-Wilbraham, Connersville, Ind. Data sheet on gas pumps for handling gases and vapors, with gas chamber entirely separated from the bearings by using stuffing box around each shaft as it passes out of the casing.

Pumps: DeLaval Steam Turbine Company, Trenton, N. J. 8-p catalog of rotary displaecment pumps to operate at motor or turbine speeds, capacities 4 to 460 gpm, pressures to 200 lb (also supplied for pressures in excess of 500 lb).

Pumps: Westco-Chippewa Pump Co., Davenport, Ia. 4-p pamphlet telling features of pumps for circulating brine or ice water, booster service, boiler feed, etc., in buildings.

Pumps: Ames Pump Company, Inc., 30 Church St., New York City. 8-p folder containing description, construction, dimensions, ratings and specifications for vertical single condensation pumps.

Recorders: General Electric Company, Schenectady, N. Y. 4-p folder describing photoelectric instrument for accurately recording any measurable quantity, including temperatures, currents, voltages, noise intensity, etc.

Rubber: The B. F. Goodrich Rubber Co., Akron, O. 4-p insert on a soft, black elastic rubber for protecting surfaces from abrasion of wet or dry materials, including lining pipe.

Sheet Metal: Ingersoll Steel & Disc Company, 310 S. Michigan Ave., Chicago, Ill. 4-p folder on stainless-clad steel.

Stokers: Detroit Stoker Company, General Motors Bldg., Detroit, Mich. 12-p bulletin on multiple-retort stokers, containing a complete description of their features.

Stokers: The Patterson Foundry & Machine Company, East Liverpool, O. 8-p bulletin describing the features of underfeed heavy-duty stokers and 4-p pamphlet on standard-type stokers.

Systems for Combustion Control: L. J. Wing Mfg. Co., 154 W. 14th St., New York City. 8-p booklet describing operation and applications of combustion control systems for burning small anthracite coals. A number of typical jobs are described and engineering data are given.

Turbines: General Electric Company, Schenectady, N. Y. 16-p catalog of steam turbines for mechanical drive of pumps, fans, compressors, blowers, pulverizers, stokers, paper machines, etc. illustrating and describing applications and construction.

Valves: Shand & Jurs Sales Co., Denver, Colo. Data sheet giving dimensions and features of quick-opening gate valves of new design.

Wrought Iron: A. M. Byers Company, Pittsburgh, Pa. 4-p bulletin describing genuine wrought-iron forging billets. 4-p folder on wrought-iron plates—differentials and extras. Cards on differentials and extras for wrought-iron galvanized and hot-rolled sheets. 4-p pamphlet, recommended specifications for refined wrought-iron bars.

Controlling Motors for

. . Temperature Control Systems
. . . . . Compressors
. . . . . Oil Burners
. . . . . Stokers

By Samuel R. Lewis*

Almost any desired operating result can be achieved by the application of the proper type of control to motors driving heating and air-conditioning equipment. Indicative of the flexibility and applications of modern control equipment have been the two previous articles in this series on control of motors driving ventilating fans and control of motors driving pumps. The present article concerns electric temperature control, oil-burner motor control, stoker motor control, compressor control, unit-heater fan-motor control, etc.

ELECTRICITY is useful in the field of heating, ventilating and air conditioning for operating temperature control apparatus, and for driving refrigerating compressors, unit-heater fans, oil burners, stokers, etc. (Applications of electrical control equipment for motors driving fans and pumps were described in November and December, 1932.) Reliable and effective control for all of this electrical apparatus is vital to operating success.

Operation of an Electric Temperature Control System

One form of electric temperature control system employs a single-phase fractional horsepower motor to operate the valve through a 1600:1 gear ratio speed reducer. The motor rotates in the same direction for both opening and closing of the valve, the reversing of the valve stem movement being accomplished mechanically at the end of each half-cycle of a cam rotated by the motor. Fig. 1 shows the electrical circuit employed.

A decrease of temperature will cause the thermostat to make contact with line *a*. If the motor valve should be closed, the switch, guided mechanically in harmony with the valve position, would be in contact with line *a* and the motor circuit would be completed. The motor then would be caused to operate to open the valve. Near the end of the full opening cycle, or when the valve is almost wide open, the switch would be thrown mechanically by the motor to contact with line *b*. This, of course, will break the motor circuit and will cause the motor to stop with the valve in open position.

When increase in temperature has caused the thermostat to contact line *b*, the motor circuit will be completed through the switch, which was left in contact with line *b*, and the motor will operate to close the valve, placing the switch again in contact with line *a*, breaking the motor circuit and stopping the motor at the closed position of the valve.

The contactors of the switch, however, are connected to both lines *a* and *b* until the end of the full opening or full closing cycle. This prevents the motor from stopping in a partly opened position, should the thermostat change contact with the original line during the time the motor is running. Thus, if the thermostat should change contact from *a* to *b* after originally closing *a* circuit, the motor would complete the cycle demanded by the *a* contact, and would continue to run until satisfying the cycle demanded by the changed *b* contact.

No form of overload or under voltage protection is necessary for the motors used to operate the valves of this system. The work required to open or close a valve, with the leverage and speed reduction employed with this system, involves only about one-half the capacity of the motor used, so that the danger of overloading the motor is small.

Should the valve be stopped from closing completely by a piece of scale on the seat, or for some other reason, a spring steel member allows the motor to complete its cycle without danger of overloading.

With this type of motor any reduction of line voltage will cause a reduction in the motor speed. Uniform speed of movement is not a factor in the operation of the valve which this type of motor controls usually, and so no objection comes on this score.

A fused line switch on the incoming power lines is the only circuit protection.

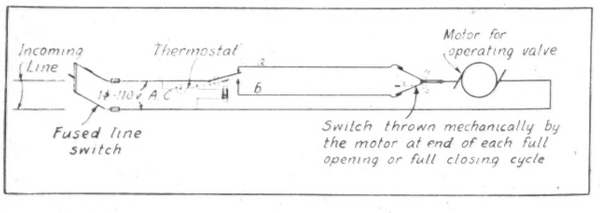

Incoming Line
Thermostat
Motor for operating valve
Fused line switch
Switch thrown mechanically by the motor at end of each full opening or full closing cycle

Fig. 1—Wiring diagram for a simple electric automatic control circuit. The motor does not reverse its direction of rotation and only the current which actuates the relay goes through the thermostat

*Consulting Engineer, Chicago, Ill. Member of Board of Consulting and Contributing Editors.

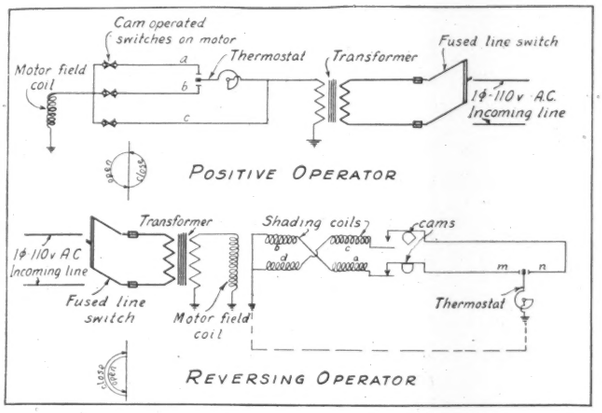


Fig. 2—Two circuits for automatic temperature control. The upper diagram shows an arrangement by which the starting current only for the valve- or damper-moving electric motor passes through the thermostat. The lower diagram illustrates a motor-reversing arrangement to achieve slowly-changing temperatures

Another form of electric temperature-control system employs 25-volt current to operate electric motors of the shading-ring induction type which move valves or dampers. These motors may be reversing or unidirectional. A diagram of the electrical circuit employed with this type of system is shown in Fig. 2.

Referring to the positive operator circuit, this is for an electric motor which operates only in one direction and which assumes no intermediate position between full closed and full open. This operates as follows:

The cam-operated switches *a* and *b* operate alternatively with each other and both operate alternatively with the cam-operated switch *c*. At the end of any cycle the motor mechanically opens switch *c* and closes switches *a* or *b*. The thermostat reacting to temperature may then close the *b* circuit, for instance, and the motor immediately will start to operate, securing its supply of energy through the thermostat. As soon as the motor moves, however, the switch *c* will be closed and switch *b* will be opened; so that the motor from that time forward receives its energy through switch *c*, saving the thermostat from carrying the running current. At the end of the cycle again switch *c* will be opened by the motor and switch *a* will be closed in readiness against the time when the thermostat, reacting to the temperature effect of the preceding motor movement, will complete closure of the *a* circuit. When the *a* circuit is closed by the thermostat the whole performance will be repeated, but this time the *b* switch alone will be closed at the end of the motor movement.

One side of the transformer secondary coil winding and one side of the motor field coil are grounded to complete the low-voltage circuit. This type of control causes complete closing or opening of the valve or damper at every cycle of operation.

Referring to the circuit of the reversing operator of Fig. 2:

Closing of the circuit by the thermostat permits the flow of current through the path *m* and through one set of shading coils, *a-b*. This induces a definite flow of current through the motor field coil, producing a definite motor rotation and valve or damper movement. At the limit of travel of the valve or damper, a set of cams breaks the motor circuit and places an opposed set of shading coils in line to be energized, whenever the thermostat, due to receiving an opposite temperature impulse, completes the circuit *n*, through coils *c-d*. This induces a reversal of flow of energy in the motor field coil and correspondingly reverses the motor rotation and the valve or damper travel.

No form of under-voltage or overload protection usually is installed for the motors of this system. The value of the current of a stalled shading ring induction motor is less than the value of the current drawn when the motor is running, so a motor never can burn out.

A fused line switch on the incoming lines to the transformer primary coil forms the only electrical protection on this system.

Control of Compressor Motors

Because of the large size motors required for the refrigerating compressor drive for central air-conditioning plants, and the steady nature of the load, this type of work is commonly handled by synchronous motors. Synchronous motors as a rule are less costly than induction motors of the same power capacity and have the further advantage of improving the power factor of the connected load.

Refrigerating compressors may be driven by low-speed direct-connected synchronous motors or by moderate speed belted synchronous motors. Separate direct-current motor-generator sets usually are employed for furnishing field excitation with low-speed synchronous motors, whereas direct-connected exciters usually are employed with moderate- and high-speed synchronous motors.

Low-speed synchronous electric motors usually may be started under full voltage, but reduced voltage arrangements commonly are provided for high-speed synchronous motors.

Starting apparatus for synchronous motors should be of the full-automatic type so as to eliminate as far as possible the personal element with its attendant danger of damage to a costly machine. For instance, the improper or untimely application of the field excitation might be disastrous to a synchronous motor.

It also is necessary that overload and under-voltage protection be provided on the controllers of synchronous electric motors.

Fig. 3 illustrates an application of a low-speed synchronous motor to a carbon dioxide compressor. The controller and field-current motor-generator set are shown at the left of the picture. On the control panel may be identified:

1. The line ammeter which indicates the load on the motor at all times.
2. The pilot light which shows when the line voltage is on.
3. Field ammeter which indicates the excitation of the motor.
4. Hand wheel on the field rheostat.
5. Start-stop push button station.
6. Temperature overload relay.
7. Field contactor operated by a field actuating relay, which applies excitation at the proper moment for synchronizing and removes excitation whenever the motor is stopped.
8. Line contactor which connects the motor to the line at starting, for full-voltage starting.

It may be desirable to start and stop refrigerating compressors by a temperature switch or by a pressure switch in lieu of the hand-operated push-button station. In such a case, both the automatic actuating device and the manual switch may be installed with a selective double-throw switch for alternative operation.

An Oil-Burner Motor Control

Oil-burner motors usually are of comparatively small size and most of them are of the induction type. Definite control of the oil-burner motor is vital to the successful and safe operation of the heating plant.

An oil-burner motor should be caused to respond automatically to:

1. Temperature or steam pressure or water level fluctuations between predetermined limits.
2. Ignition failure.
3. Combustion failure, either because of fuel shortage or because of temporary fuel stoppage.

The wiring diagram (as used in one commercial circuit) to the controllers required to gain these effects is shown in Fig. 4 for an induction current.

The operation is as follows: When thermostat *T1* calls for heat (this also may be a pressurestat calling for increased steam pressure) a circuit is closed through a low-voltage transformer, energizing the pilot relay, which, in turn, closes a line voltage circuit through the coil of relay *1*. Relay *1* closes the oil-burner motor circuit to the line through a magnetic starting switch which is provided with a thermal overload protector, and relay *1* also supplies line voltage to the ignition device which may increase the opening of a magnetic gas valve on the pilot light, or which may operate a spark coil.

Also built into relay *1* is a thermal switch *S1*, consisting of a thermostatically-sensitive metallic strip operating a latch, which in the cold position maintains a contact for the circuit of relay *1*. Under this metallic strip is a resistance coil, connected in series with the coil of relay *1*.

As soon as thermostat *T1* functions to start the motor through the process described, the line current passing through the resistance coil generates heat and causes the thermostatically-sensitive metallic blade to bend.

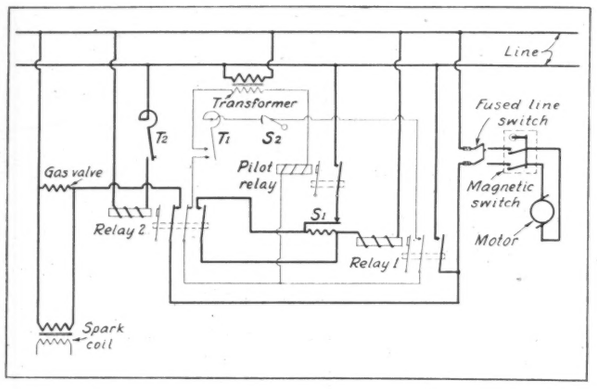


Fig. 4—An oil burner control circuit. Thermostat *T1* responds to the room temperature and starts the ignition circuit and the pump-fan-motor circuit through relay *1*. Thermostat *T2* in the smoke pipe must be kept hot or it will stop the foregoing mechanism. If thermostat *T2* over-operates it must be reset manually. Switch *S2* is operated by too high or too low water in the boiler or by too high a temperature in the furnace jacket

If, in the meantime, combustion has started at the burner, the resulting hot gases of combustion will pass over a thermostat *T2* placed in the smoke pipe or its equivalent. The heat from the burning oil will influence thermostat *T2* and will close a circuit energizing relay *2*. When relay *2* is energized, it closes a circuit shunting out the resistance coil of the thermal switch *S1*, thereby preventing the breaking of the oil-burner motor circuit.

Conversely, if combustion has not taken place within a few seconds after the oil-burner motor starts by the operation of thermostat *T1*, thermostat *T2* will remain cold and will permit the line current to pass through the resistance element of the thermal cut-out switch *S1* until the heat generated expands the metallic strip and breaks the oil-burner motor circuit.

The thermal cut-out switch *S1*, once released, must be manually reset before the burner can be started.

If, after the burner has been in normal operation, there should occur failure of combustion due to a temporary oil stoppage, thermostat *T2* will respond quickly and will open relay *2*. The starting circuit then is re-established to provide ignition in case the oil supply should return before the time that the thermal cut-out switch *S1* could function to break the oil-burner motor circuit.

Fig. 3—Synchronous motor direct connected to a refrigerating compressor used in an air-conditioning plant. The exciter and the starting panel are shown at the left

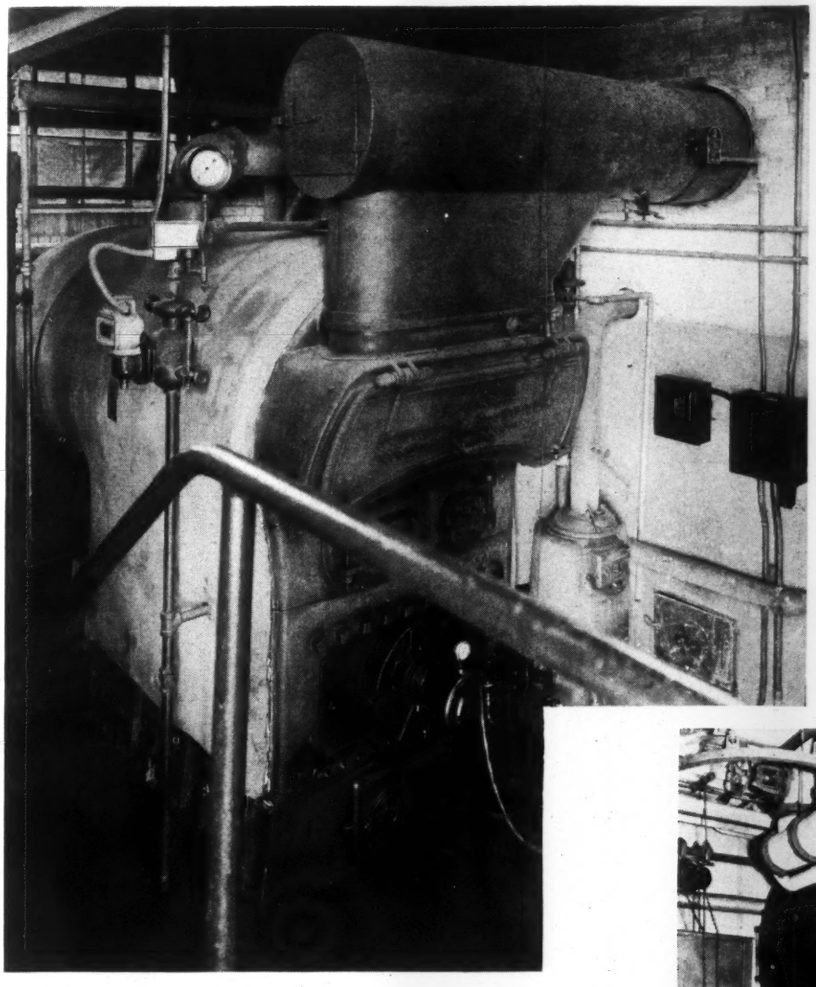
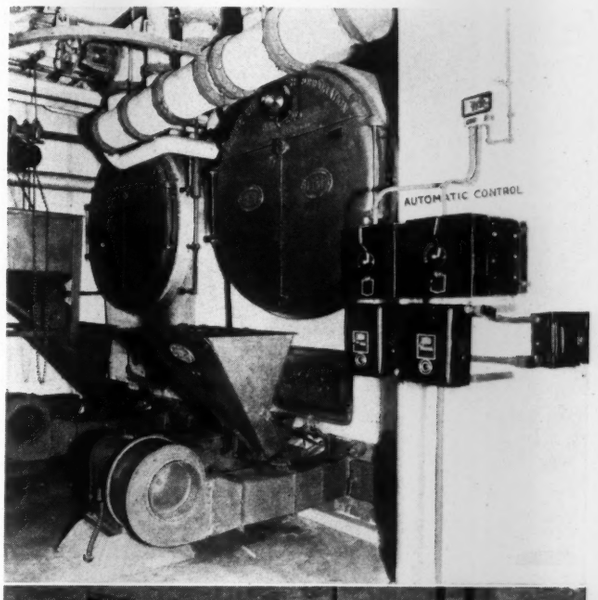
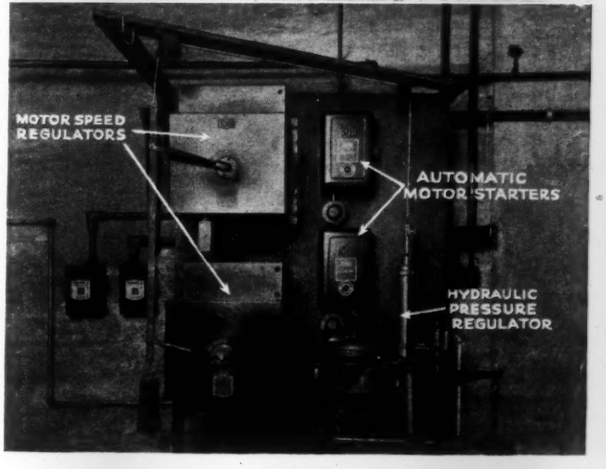


Fig. 5 (Left)—An oil burner, provided with the control illustrated in Fig. 4, installed in a low-pressure heating boiler. *T1*, a pressurestat, is shown near the steam gage. *S2* is the high-low water cutout switch mounted on the water column. *T2* is a thermostat shown mounted in the breeching. The fused line switch and magnetic switch with push-button starter are shown mounted on the wall of the stack

Fig. 6 (Below) — Manual-automatic stoker control. The stoker speed is set by the levers on the large control cabinets. The small enclosed switch above these cabinets stops and starts the stokers through the magnetic starters under the large cabinets. At the right are the cutout switches

Fig. 7 (At Bottom)—Variable speed control for automatic stokers. The hydraulic pressure regulator responds to very slight variations in steam pressure and operates the electric motor regulators to compensate





In series with the thermostat *T1* (or the pressurestat) is a float switch *S2* operating from the water line in the boiler. In the event of dangerously low or high water, the primary control circuit will be opened and the oil-burner motor will stop.

Controlling Stoker Motors

Automatic electric-motor-driven coal stokers for use with heating boilers generally are provided with means of control which are actuated by fluctuations in steam pressure or temperature.

The majority of the stokers which are installed under moderate-sized heating boilers are provided with automatic start-stop control actuated by steam-pressure fluctuations, with a manually pre-set controller for governing the speed at which the coal feeding mechanism operates. Larger sized installations may be equipped with fully automatic speed-regulating controllers. Motors adapted to this service are high starting torque, wound rotor, induction type.

Fig. 6 illustrates the control equipment for the two stokers shown. This installation is of the hand-speed-set, automatic start-stop type. At the extreme right may be seen the fused safety switches on the incoming lines.

The large cabinets with hand levers on their fronts

are the armature resistance rheostats for hand-setting of the speed. Below the resistor cabinets are located the magnetic starting switch cabinets with the overload and low-voltage relays. The steam-pressure-actuated switch which closes the line to the starters is shown above the resistor cabinets.

Fig. 7 illustrates fully automatic speed regulating controllers.

Fluctuations in steam pressure are transmitted to a diaphragm on the hydraulic pressure regulator. This causes movement of a piston in the cylinder, which motion is transmitted by means of the cable and pulleys and counterweight to the levers on the motor speed regulators, which produce a corresponding speed of the stoker motors. The fused safety-type disconnect switches on the incoming power lines may be identified on the wall to the left of the control panel.

Controlling Unit-Heater Fan Motors

If the fans installed in connection with recirculating unit heaters are stopped and started automatically from a thermostatic electric switch located on the return piping side of the unit heater, and the thermostat which controls the room temperature used only to govern the admission of steam to the blast coils of the unit heater the possibility of operating the fan and hence blowing cold air and causing draft complaint before steam has warmed the heating coils is precluded.

The motors usually furnished for such fans are of the single-phase induction type. The starters for these motors generally consist of single-pole switches which throw the motor across the full voltage of the power line. A thermal overload switch always should be used in series with the starting switch for protection of the motor.

Under-voltage protection usually is not provided for the small-size motors of unit heaters or unit ventilating machines.

Speed regulation of the fans of a unit heater may be desirable as a means of controlling the output. This may be accomplished by inserting a manually-operated speed controller in series with the motor. This controller usually is provided with two or three standard speed settings.

Protection of Motors

In general, motors should be equipped with the following protection:

1. Every motor should have a disconnect switch which will open all power lines leading to the motor and which will disconnect the control.

2. Every motor should have a device to afford protection from overheating the motor windings by excess current, such as would occur in case of a heavy motor overload. This consists usually of a relay so constructed as to break the line circuit in case of a continued overload, thereby stopping the motor. The resetting of these devices must be done by hand, purposely, so that personal attention will be given to any condition serious enough to cause the relay to function.

Ordinary fuses do not afford protection equal to that provided by thermal relays and are not satisfactory for overload service, although they are used occasionally for very small direct-current motors which do not draw heavy starting current. While there is an inherent time-lag to a fuse, it is not of the same order as the heating of a motor winding. Thermal relays will not trip instantaneously, therefore, but will function only when a heavy current is sustained long enough to cause dangerous overheating. Fuses are useful however, for protection of motor feeders against grounds, short circuits, and similar troubles in the line.

Recently, miniature thermally-operated circuit breakers, possessing the same heating characteristics as ordinary copper line wire, have been made available for the same services as fuses. With this new type of protection the trip can be reset by hand, and the circuit thus will be re-established.

3. All motors except the very small fractional horsepower ones should have under-voltage protective devices to stop them in case of low voltage.

Units Condition Concrete Testing Laboratory

AN unusual application of air conditioning at the Bureau of Reclamation's concrete testing laboratories in Denver aids in the securing of information and data on the properties of concrete before completing the final design for Hoover Dam, Las Vegas, Nevada.

In each of the two laboratories—one in the federal building, and the other in a rented building—refrigeration and air conditioning systems automatically maintain temperatures and humidities which simulate conditions encountered on the construction work.

In the federal building laboratory, there are four concrete curing rooms; the other laboratory has one curing room.

In a large mass of concrete such as will be poured in the Hoover Dam, the concrete will remain practically saturated until it has set so the test specimens will be cured under conditions which approach as nearly as possible these conditions. The concrete curing rooms can be maintained at a temperature of 70 F and a relative humidity of 100 per cent in addition to standard laboratory practice. In the federal building laboratory, three rooms are maintained at 70 F and 100 per cent humidity and the fourth at 70 F and any humidity between 50 and 100 per cent. The allowable variation in temperature is 2 F and 3 per cent in humidity.

Removing Heat Generated by Cement

Tests on existing dams have proved that it takes a long time to dissipate the heat generated by the cement. In some dam test tunnels temperatures well above 100 F have been encountered several years after completion of the dam. As concrete will continue to contract until all the heat has been removed, it is feared the grouting between the various sections will fail unless practically all of the heat is removed before applying the grouting. It has been decided to resort to refrigerating the entire

mass of concrete to be poured for Hoover Dam to do this. Calculations show it will require approximately 1,000 tons of refrigeration per 24 hours to do the job. The plan is to install pipes in the concrete as it is poured and to circulate cold water through them until the heat is removed.

Control of Conditions in the Laboratories

In the federal building laboratory, three rooms require cooling, heating and humidifying and the fourth also requires dehumidifying. The automatic control system must operate in all seasons and with all variations

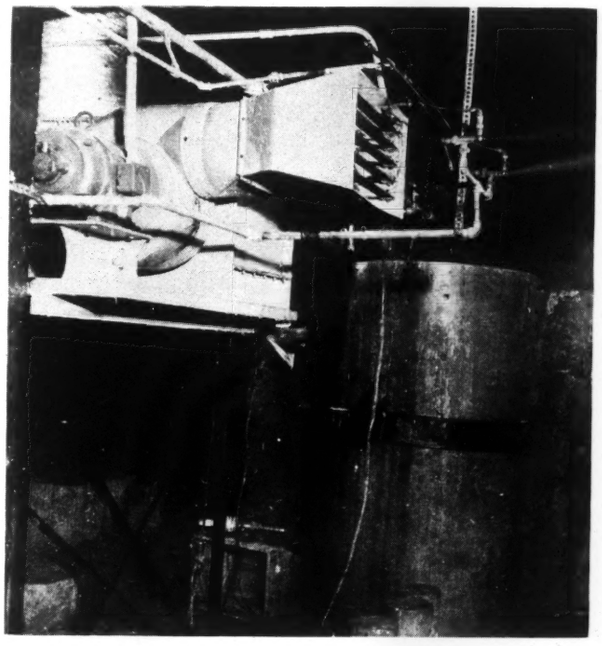
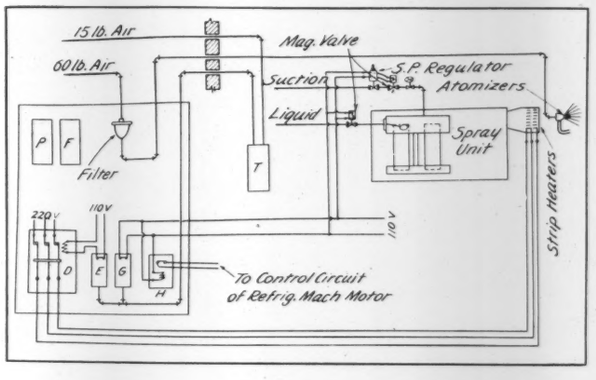


Fig. 1 (Above)—Air-conditioning unit, electric strip heaters and atomizers which maintain temperature within a 2-degree variation and relative humidity within a 3-per cent variation in one of the concrete curing rooms established to obtain data for use in the design of Hoover Dam. Fig. 2 (Right)—Control diagram for air-conditioning unit which maintains room temperature and humidity within close limits; operation is explained in the text*

in room heat loads without adjustment. The refrigerating equipment consists of a standard self-contained unit operating on float-controlled air-washer coils in each room. Each coil is equipped with a suction-pressure regulating valve and a magnetic liquid valve; coils in two of the rooms also have magnetic suction-line valves. Heating is effected by means of electric strip heaters in the outlets of the air-conditioning units. The special ceiling-type spray units include a complete assembly of air washer, refrigerant coils and control, electric heaters, pump and fan. Humidification is accomplished by compressed-air type atomizing nozzles.

In the rooms maintained at 70 F and 100 per cent relative humidity the suction pressure regulators are set to maintain 70 lb pressure in the refrigerant coils; in the other room the suction pressure is varied to give the desired dew-point for the condition to be maintained. There is a suction pressure switch at the machine which controls the two-speed motor starting switch. When the suction pressure is above 40 lb at the machine, it will operate at high speed and when the suction pressure at the machine drops to 40 lb or below, it will operate at low speed.

Fig. 2 shows the automatic control. *P* is a hand starter controlling the motor driving the pump on the spray-type unit; *F* is a hand starter controlling the fan motor. While the room is in use the pump and fan are operated continuously. The automatic devices are operated with compressed air at 15 lb pressure (the air compressor also supplies air at 60 lb to the atomizers). With room temperature of 70 F, thermostat *T* will maintain approximately 5-lb pressure in the control line. With rising temperature in the room, the thermostat increases pressure in the control line; at 7 lb, switch *G* closes circuit and opens magnetic valves in suction and liquid lines to the air-conditioning unit. At the same time motor control relay *H* closes and calls for refrigeration. As the room is cooled down to 70, the pressure in control line drops to 5 lb and switch *G* opens circuit, closes magnetic valves and opens the refrigerating-machine circuit. If the room temperature drops below 70, the pressure in the control line drops and at 3 lb pressure switch *E* closes circuit and through magnetic switch *D* turns on the strip heaters. As the temperature rises, the pressure in the control line rises and at 5 lb, switch *E* opens circuit and shuts off heat. With four rooms, any one requiring refrigeration will start the compressor through relay *H*, which will operate as long as a single



room needs cooling. When only one or two rooms need refrigeration, the compressor operates on low speed.

In the optional humidity room the automatic control equipment is the same as shown in Fig. 2 and the operation will be the same when this room is maintained at 100 per cent humidity. When a humidity less than 100 is desired, additional controls are required, with one set for 50-70 per cent and another for 70-95 per cent. A dew-point or a wet-bulb thermostat controls the refrigeration and the dry-bulb thermostat controls the strip heaters.

*Illustrations from A. K. Snyder, York Ice Machinery Corporation, York, Pa.

INTERESTING Equipment..... Developments

A method and the equipment for spraying a coat of metal to rust- and corrosion-proof almost any surface is being introduced in this country by the Metallizing Company of America, Inc., Los Angeles, Calif. The metal to be sprayed is fed into the gun in the form of wire. In the gun it is melted and sprayed like paint under high pressure. The coating may be built up as thickly as is desired, the connecting bond being due to the heat and the pressure. Among the metals that can be sprayed are zinc, nickel, lead, bronze, copper, monel, aluminum, brass and tin.

This process may be used in re-galvanizing or protecting welds in piping, and for repairing or protecting tanks used in process work, etc. One interesting application was the coating of a stock blower with lead to protect it from the acid fumes.

Railroad Car Cooling Units

The Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa., now builds complete railroad car cooling apparatus. This equipment has been developed to embody simplicity, reliability and efficiency.

The units consist of three major parts:

The evaporator and air circulation unit mounted under the car roof and used for cooling and dehumidifying the air circulated in the car.

The condenser unit, mounted under the car body and consisting of an air-cooled condenser, a motor-driven compressor, a cooling fan, and the control.

The 15-kw generator (usually truck mounted) driven from the axle. This unit supplies power for the air-conditioning equipment and for car lighting and charging the battery.

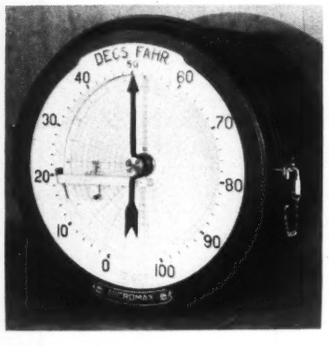
The standard car-cooling equipment is rated at 6 tons of refrigeration.

New Indicating Recorders

Among instrument displays at the recent power show were four new round-chart micromax indicating recorders made by Leeds & Northrup Company, Philadelphia, Pa. One was recording CO_2 in the combustion range, one was recording pressure over a range of 0-200 lb per sq in., the third instrument was recording room temperatures and the fourth was recording temperatures in the range of 400-900 F.

This instrument has a boldly-lettered circular scale that can be read across a large room. It uses a 24-hour circular chart with straight-line time coordinates instead of curved ones for readability and is one of the few round-chart recorders with such coordinates. The calibrated portion of the chart is 3½ in. wide, and the chart is 10⅜ in. in diameter overall. The recorder is made as a single-point instrument only, and can be equipped with two signal or control contacts. Potentiometers are manually standardized. The mechanism is mounted on a frame that is hinged to the case and can be swung out of the case if desired. The case is 16½ in. diameter, and may be mounted either flush or hanging on the face of a wall or panel as specified.

The temperature detector (thermocouple or resistance thermometer) may be replaced when necessary without sending the recorder back to the factory; it may be installed at any distance from the indicating recorder or shifted from place to place at will. This instrument is also especially adapted to the recording of temperatures across the entire ranges of thermocouples, of resistance thermometers, to the recording of smoke density, liquid levels, CO_2, speed, solar radiation and many other applications.



A new multi-purpose case for this company's micromax recorder has also been announced. This case is dust-tight, and vapor-tight when desired. It can be supplied for mounting in any position as specified — either flush in panel, on front of panel or wall, or on table or bench. The case is sealed by simply turning the handle, thus pressing the door tightly against a rubber gasket.

Announces a Line of Stokers

The Patterson Foundry & Machine Company, East Liverpool, O., has recently announced a new line of stokers designed for large heating or small process steam loads.

In the standard hydraulic model, unit type, the motor is placed on top of an oil reservoir, which acts as a supply and overflow chamber. A small rotary pump, driven by V-belt from the motor, submerged in this reservoir, provides pressure to operate the hydraulic ram piston, through a slide valve and regulating valve. By means of the regulating valve it is possible to get any desired

speed on the ram. In addition, an adjustment is provided on the distributing rod so that the distribution of coal in the retort can be varied at will from outside. A relief valve is located in the oil reservoir to add additional protection against overloads in case motor controls fail.

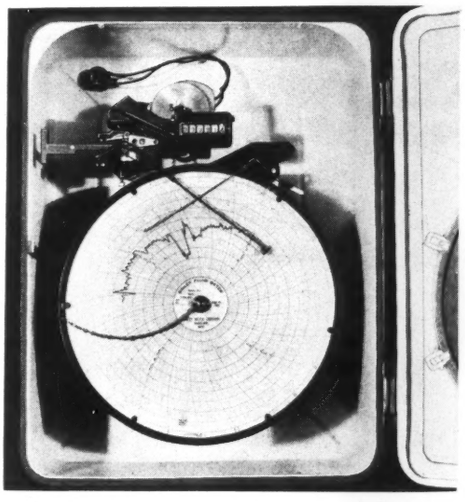
This machine can be furnished with the fan V-belt driven from the same motor that drives the pump, or with a separate motor-driven fan located away from the boiler front, in the latter case controls being furnished to permit of either synchronous or separate operation. In addition to the usual controls, a machine limit switch is provided which stops the machine with the ram in an advanced position, thus closing off all communication to the furnace when the stoker is down and preventing burning back to the hopper.

The grate area is made up of sloping, slot-type tuyeres and slotted side dump bars, with the plenum chamber underneath divided into zones with damper control. Normally no air is admitted under the dump bars, but at times, perhaps when preparing to shut down, it is desirable to provide air at that point to accelerate burning of any fuel that may remain in the clinker or ash, and this is possible by opening the dampers. This same stoker is available with gear drive, in which case a chain drives a crankshaft to give motion to the ram.

New Integrator for Meters

A development incorporated in the new line of fluid meters made by Bailey Meter Company, Cleveland, O., and recently announced is the escapement-type integrator, which operates in a simple, but accurate manner. The general appearance of this integrator is illustrated in the view of the meter interior; it will be noted that the total flow in gallons, pounds or other units, is given on a six-figure counter.

The principle of operation is as follows: A heart-shaped cam, which has a uniform angular rise, is geared to a synchronous motor and rotated at the rate of two revolutions per minute. By means of a friction clutch between the cam and the escape wheel, the motor also drives the escape wheel and integrator counter at constant speed when the pawl is not engaged. When the pawl engages with the escape wheel, the integrator counter is held stationary, but the friction clutch allows the cam to continue rotation.

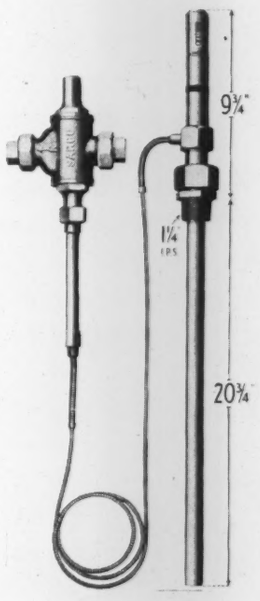


The roller arm is pivoted near its left end to the flow mechanism, so that the position of the pivot varies only with changes in the rate of flow. The right end of the roller arm moves up and down under the action of the rotating cam, causing the pawl operating pin to move up and down also. It is this operating pin which is responsible for engagement and disengagement of the pawl with the escape wheel.

This escapement-type integrator is accurate as its operation is governed by the rate of flow at each engagement and disengagement of the pawl. Since the pawl engages and disengages during each revolution of the cam for all rates of flow between 0 and 100 per cent, the integrator total is adjusted to the prevailing rate of flow four times each minute.

Regulators for Industrial Heating

A new line of temperature regulators for industrial heat control (enclosed spaces, drying rooms, kilns, open kettles, vats, tanks, etc.) has been made available by Sarco Company, Inc., New York City. The regulators operate by liquid expansion; each consists of a thermostat tube, a flexible connecting tube, a valve-moving plunger, and a valve. The thermostat, connecting tubing and plunger cylinder are filled with a special oil.



Each regulator may be calibrated at any point within the range of 0-300 F. Regulator valves are of a double seated balanced type. The valve moving plunger and displacement piston for adjusting the instrument have packless seals of corrugated seamless bronze tubing.

These instruments are designed for throttling control. Where a wide open valve is desired until the calibrated temperature is reached, when the valve must shut and remain closed for long periods, an indirect type of regulator is available. The thermostat operates a small pilot valve which allows steam pressure to reach a large diaphragm, to open the main valve.

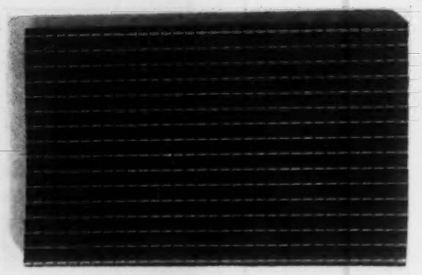
Pipe Covering for 1800 F

A new pipe covering—a product of the Geo. B. Smith Chemical Works, Inc., Springfield, Ill.—is adapted for use in the ordinary range of temperatures and in addition may be used at temperatures as high as 1800 F. The covering is not soluble in water or moisture. It is made

of aluminum magnesium silicate and long-fibered asbestos and covered with a thickly woven fabric for durability and appearance. The pipe covering is furnished in all standard pipe sizes in standard and double standard thicknesses.

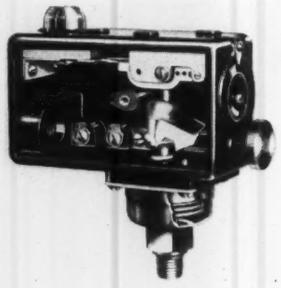
Belt Surface is Treaded

Illustrated here is the surface of a new leather belt which has been introduced by E. F. Houghton & Co., Philadelphia, Pa. It has a non-skid surface (produced



by pressing down or indenting part of the surface of the belt) to concentrate the pressure between the belt and the pulley on the ribbed tread. Application of this type of belt to short-center drives is stressed particularly by the manufacturers.

Two New Control Switches



Two new control switches have been announced by Detroit Lubricator Company, Detroit, Mich.—one is furnished in various models for the control of temperature, pressure, or vacuum. The other is a clamp-on type for temperature and operates by surface contact with the metal.

In the first-mentioned switch (illustrated here with cover removed) a bellows element actuated by pressure direct or by a volatile liquid for temperature operates the switch mechanism to make or break a circuit. A permanent magnet at the contacts provides a quick make and break and prevents excessive arcing. All models are enclosed in a black bakelite case and both operating range and differential are adjustable.

Double Glass with Air Space

An insulating and frost- and moisture-resisting glass has been developed by Charles D. Haven, Milwaukee, Wis. It consists of a double glazing with a sealed space between which contains dehydrated air. Sealing is done by a special process and in a manner to permit sufficient flexibility so that the glass will not be broken because of unequal expansion or contraction of the two panes. Air spaces vary from ⅛ to ½ in. It is used in windows and door openings in place of a single sheet of glass.

According to tests made at the University of Illinois (as reported by the manufacturer), with 0 F outside, a single pane of glass had an inside temperature of 2 F, with double-sash the inside surface temperature was 17 F and with the insulating double glass it was 40 F. Heat transmission is given as 0.45 Btu per hr per sq ft per degree difference in temperature.

As an example of the application of this insulating glass, sheets 66 by 40 with ½-in. air space are being used at the zoological gardens near Chicago in a house for tropical birds in which the humidity is high.

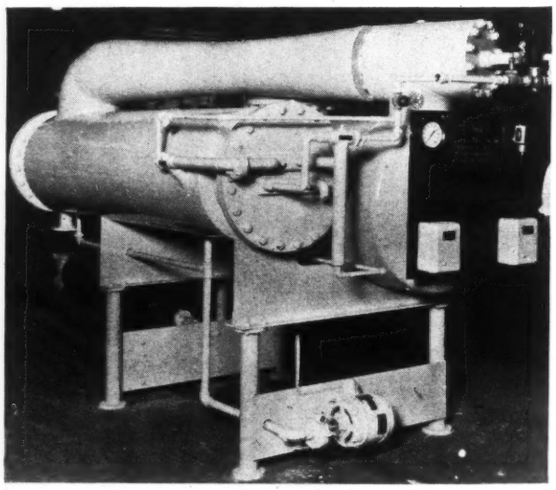
Simplified Draft Gage

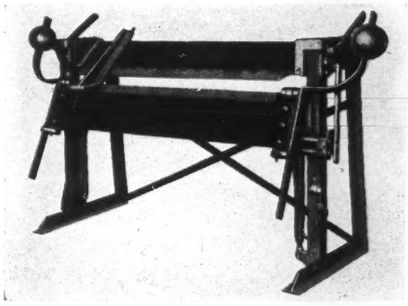
A "smoke-tube" draft gage has been announced by E. Vernon Hill Co., Chicago, Ill., for taking draft readings, static pressure readings in ducts, etc. It consists of a length of rubber tubing, an intake nozzle and a calibrated glass scale which is filled with smoke from a cigar or cigarette. The flow of air through the intake nozzle into the glass tube forms a jet which clears the smoke from the upper portion of the glass tube, the length of the jet of air depending on the pressure. The "height" of the smoke in the tube enables reading of the scale, which is calibrated against a master gage by test and reads directly in inches of water.

25-Ton Steam-Jet Unit

The illustration shows a steam-jet refrigeration unit which has lately been shipped by Ross Heater & Mfg. Co., Inc., Buffalo, N. Y. This unit is suitable for cooling water for air conditioning, cooling drinking water or for supplying water for industrial processes at temperatures between 35 and 60 F. The unit shown operates on a steam pressure of 100 lb gage; its capacity (at 40 F) is 25 tons.

Units of this type are made to operate on steam pressures from atmospheric up; the unit shown is complete and requires no special foundations. This company is now at work on a 300-ton unit for Building No. 3 of Radio City in New York, according to an announcement.





A New Metal Brake

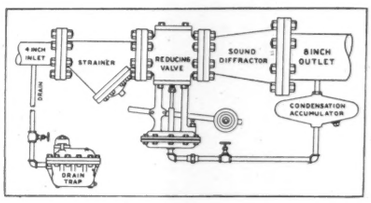
A new metal brake which is now being used in the construction of the new post office at Rockford, Ill., is announced by Whitney Metal Tool Company of that city.

The features of the brake are that it operates the reverse from other machines of its kind. The platen which holds the work (*i.e.*, the part which the material is held against) is stationary or solid, enabling the brake to make very square corners. The lower portion or clamping rail and bending apron slides on the two end posts. Two coil springs balance the weight of the lower mechanism so the operator is relieved of doing any lifting.

The second feature is the fact it is a dual machine, as in ten or fifteen minutes it can be converted into a box or pan brake as well as a bending brake.

Reduces Steam Noise

Application of a sound diffractor for use in new or existing installations — recently made available by Boylston Steam Specialty Co., Chicago, Ill.—is shown in the sketch here.



It is applied to the outlet flange of a steam pressure reducing valve to prevent noise caused by the steam due to its increased velocity after it passes through the reducing valve. Diffractor discs are spaced in the device to repel the propagation of vibrations.

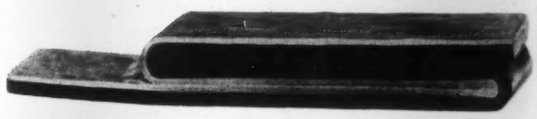
Corkboard is Moisture-proofed

A new type of corkboard for cold-storage rooms (Armstrong Cork & Insulation Company, Lancaster, Pa.) is coated on both faces of the board at the factory with a special mastic compound applied mechanically under pressure to prevent air and moisture infiltration.

Moisture "short-circuits" insulation, causing it to lose some of its efficiency. Tests show that one per cent of moisture in insulation means five per cent loss of efficiency.

Two-Ply Steel Illustrated

In a previous issue the development of a combination metal having a thin surface of stainless steel, the greater proportion and weight consisting of an ordinary steel



under- or back-surface was briefly mentioned. It is made by the Ingersoll Steel and Disc Company, (Division of the Borg-Warner Corporation) Chicago. The stainless steel is welded in a composite ingot to a mild steel foundation by a patented process. The photo reproduced here shows how the bond remains uninterrupted even in the case of a very sharp bend.

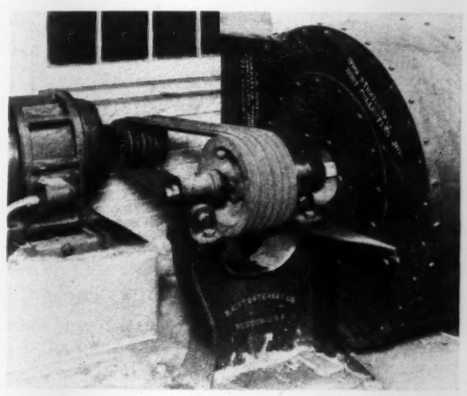
This stainless clad material has been used for hoods, stacks and duct work to carry fumes from large soap kettles by one of the major producers of soap and cleaning materials. Where corrosive conditions are severe, its use may be indicated.

The stainless clad steel may be fabricated with the same equipment used for shaping ordinary steel, as its stainless surface forms but a small portion of the total thickness of the sheet. It may be welded by using a stainless steel welding rod to assure the continuity of the stainless surface.

In duct work the mild steel sides of the sheets are on the outside of the duct, and may be protected from atmospheric corrosion by a coating of asphaltic, aluminum or other oxidation-resistant paints.

Announces New V-Belt

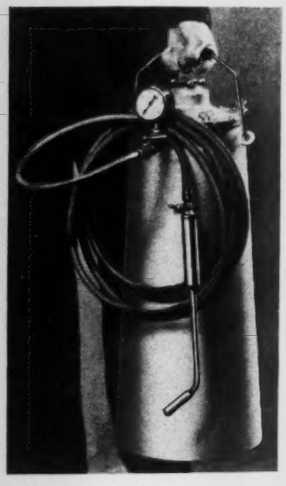
A V-belt with "no inelastic stretch" has been developed for air-conditioning equipment by The Manhattan Rubber Mfg. (Division of Raybestos-Manhattan, Inc.) In light-duty high-speed sizes, a "whipcord" construction is used, and for heavy-duty service, a more rugged "plycord" design.



The former consists of one layer of continuously wound cords imbedded between two layers of tie-gum. The balance of the belt is filled out with the right proportions of extensible and compressible rubber with a closely-woven rubberized duck jacket to complete the structure.

The latter is designed for more severe service and is made up of continuously wound strips of cord fabric with one layer of tie-gum between each ply. When built up to the right thickness and correct number of plies, it is cut to the contour of the belt.

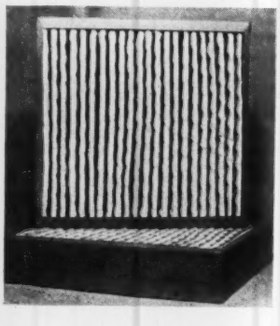
New Solder-Braze Equipment



A portable soldering and brazing equipment which has been made available by Swift Equipment Manufacturers, West Alexandria, Ohio, consists of a simplified acetylene generator, pressure gage, the necessary hose, and the torch tips and attachments. It is expected that one of its principal uses will be the soldering or brazing of de-oxidized copper tubing and slip fittings, particularly for industrial piping where hot water is maintained in the lines at all times. With this equipment an air-acetylene flame is obtained, the acetylene being generated from carbide at a pressure of about 8 lb; the tip is so designed to inject the air to make a balanced flame. It is said that the acetylene costs about 1c per cu ft when obtained in this manner.

By utilizing a small flashback check and safety blow-off, this equipment may be used for small oxy-acetylene welding and cutting jobs. The equipment has a number of miscellaneous uses in addition to those named.

Filter Has New Features



The development of a new air filter is announced by Davies Air Filter Co., New York City. It embodies the same principles that have been applied in the construction of this company's previous dry air filters. It, however, has the advantage of a very large filtering surface and a low resistance to air flow.

The distance between the folds of material is maintained constant by means of corrugated spacers (the corrugations are four inches deep) to eliminate possibility of the corrugations collapsing.

The manufacturer lists these features of the new filter—lightness, sturdiness, ease of handling, and low cost.

Scale Checks Coal

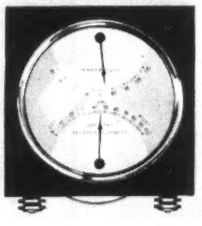
Invaluable in promoting efficiency and cutting cost is equipment designed to keep track of the sources of costs. The printing scale (Toledo Scale Co., Toledo) recently installed in The LaSalle and Koch Company department store, Toledo, Ohio, is an example.

Necessity of reducing heating costs was a factor in the decision to provide a positive check on incoming coal deliveries. The loaded truck runs onto a scale platform where the gross weight is recorded; the coal is then dumped into bins and the empty truck weighed. A printed record is made in triplicate, one copy for the coal company, one for the accounting department, and one being retained by the receiving department.

The heating plant, in addition to supplying steam for its own eight-story building, furnishes heat for several adjacent buildings. The annual coal consumption is about 8,000 tons.

Shows Temperature, Humidity

An inexpensive humidity and temperature indicator has been placed on the market by Fee and Stemwedel, Inc., Chicago, Ill. The case is modernistic black and chromium and both temperature and humidity are indicated by pointers beneath a convex crystal glass. The instrument is 4⅝ in. wide, 4¾ in. high and 1½ in. deep.



Conventions and Expositions

American Oil Burner Association: Annual convention, June 12-16, Hotel Stevens, Chicago, Ill. Secretary, H. F. Tapp, 342 Madison Ave., New York City.

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American Society for Testing Materials: Annual Meeting, June 26-30, Hotel Stevens, Chicago. Secretary, C. L. Warwick, 1315 Spruce St., Philadelphia, Pa.

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Heating and Piping Contractors National Association: Annual convention, July 25-28, Chicago. Secretary, J. C. Fitts, 50 Union Square, New York City.

American Gas Association: Annual convention, September 25-29, Chicago. Headquarters Office, 420 Lexington Ave., New York City.

Bernoulli's Theorem . . . . What It Means

By J. C. Reed and E. E. Ambrosius*

QUITE frequently old and well-established formulas in engineering practice are used without a clear conception of their derivation. Recently the authors had occasion to apply Bernoulli's theorem for the flow of fluids

$$W\left(\frac{v_1^2}{2g}+h_1+\frac{P_1}{\rho}\right)=W\left(\frac{v_2^2}{2g}+h_2+\frac{P_2}{\rho}\right)+\text{losses}$$

and the question arose as to the precise form in which Bernoulli expressed it. Some writers state it in terms of head while others state it in terms of energy. Since this theorem is the basis, directly or indirectly, for all our laws of fluid flow, it seemed worthwhile to consult its original source and then to present the results.

Bernoulli's theorem involves the principle of *vis viva* (kinetic energy as we know it today). This principle is usually stated thus: "Work done on a body = gain of kinetic energy," or, putting it in equation form:

$$F_s = \tfrac{1}{2} MV^2 - \tfrac{1}{2} MV_o^2$$

where V and V_o are the final and initial velocities.

This principle was first employed by Huygens in connection with his study of pendulums and the idea is concisely stated by Ernst Mach (p. 179, Science of Mechanics), "Since the work done in the descent is determined by the motion of the center of gravity, we have, work done in descent equal to *vis viva.*"

Leibnitz in 1695 first called the expression, mv^2, *vis viva* or *living force*. He also said, in opposition to Descartes, that this was the true measure of the force of a body in motion. Corolis modified Leibnitz's expression somewhat and stated that it was $\frac{1}{2}\ mv^2$ which should be called *vis viva*; it is now generally called kinetic energy.

This principle of *vis viva* or kinetic energy was employed by a number of men and expanded by Daniel Bernoulli so that it would provide for a greater generality, but he added little to the fundamental principle. It was known that this principle could always be employed to solve problems of moving bodies, when the total distance traversed and the forces which act upon each element were known. But Bernoulli went a step further than the others and showed that where movable bodies are in a state of mutual attraction the change of *vis viva* is determined solely by the initial and the final distances from one another and not upon the form of the path. Bernoulli applied this principle to the motion of fluids, stating it, however, in a slightly different form but with identical meaning. It was in the application of this principle to hydrodynamics that he proved that the loss of potential energy is equal to the increase of kinetic energy provided a frictionless fluid is under consideration. In *Hydrodynamicae* he deals with this principle at length.

Hydrodynamicae deals with the equilibrium, the pressure, the reaction and varied velocities of fluids, both theoretically and practically. It is a compilation of material and work of his own while at St. Petersburg. It was dedicated to Ernesto Johanni (Ernest John) Prince of Livonia. This book, his outstanding work, was finally written at Basel and published in 1738. He was indebted to the Academy of Science for all of its help and for suggestions from its members. Like all scientific works of the period, it was written in Latin.

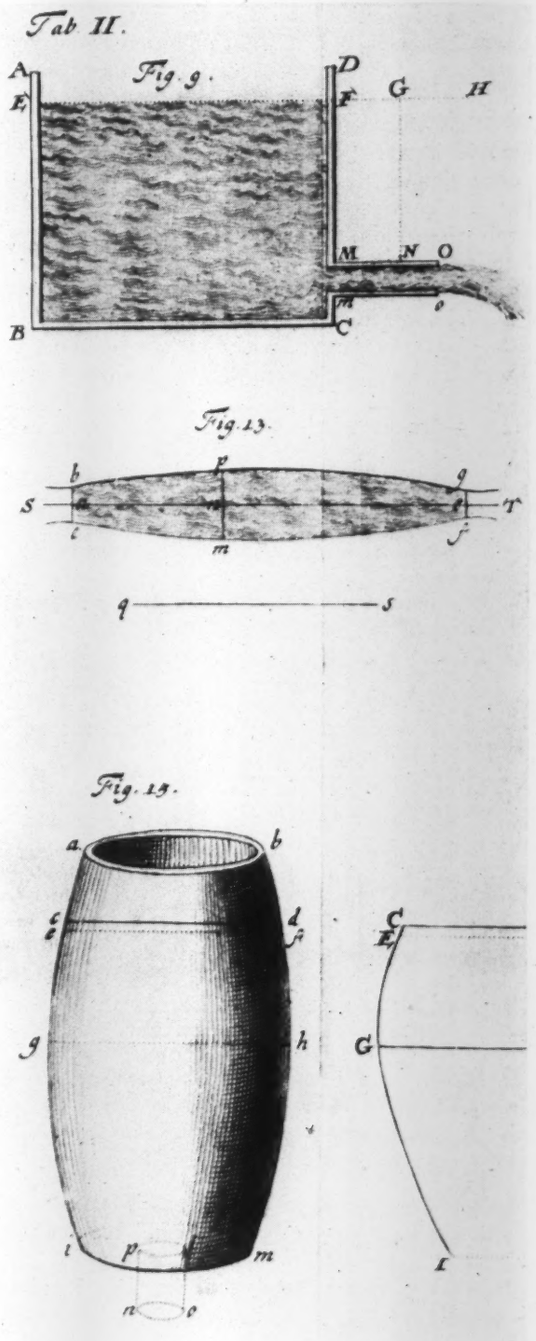

Fig 1 — Some of the diagrams used by Bernoulli in his study of the flow of fluids

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The following is quoted from this work:

"Hydrodynamics—Section Three

"Concerning the velocity of a fluid flowing from a vessel of any shape, through an opening of any kind.

"Article 1

"First before we try to determine the effect of gravity on the flow of water, let us review those principles which were so well presented in articles 18 to 22, and it is thought that we can add to these principles . . . "[1]

Article 2, Section 3, of *Hydrodynamicae* considers water flowing through a horizontal pipe of any shape (shown in Fig. 13 of Fig. 1) with the velocity known at any one section. It was desired to find the *energy* of all the water in this pipe. And he proves that the *energy* of all the water in the pipe was the same as if an equal mass was concentrated at the center of gravity. It is of prime importance to note that Bernoulli always expressed his quantities in terms of *energy.*

No doubt this work of Bernoulli could be greatly clarified if it were given a present day interpretation. The deduction of the fundamental laws of fluid motion is most easily understood with an ideal frictionless and incompressible fluid. Assume steady flow with an ideal fluid to exist, and that weight W passes at section I, in Fig. 2. Then also, the same weight W will pass section II. The kinetic energy (vis viva) at section I is equal to $\frac{1}{2} MV^2_{\text{I}}$ where $M = W/g$. The potential energy at section I due to the weight W is equal to $Wh_{\text{I}} + \frac{P_{\text{I}}W}{\rho}$. ρ = density of the fluid and P_{I} = static pressure in lb per sq ft. Therefore, the total energy at section I is equal to

$$W\left(\frac{V^2_{\text{I}}}{2g} + h_{\text{I}} + \frac{P_{\text{I}}}{\rho}\right)$$

In like manner we can find that the total energy at section II is equal to

$$W\left(\frac{V^2_{\text{II}}}{2g} + h_{\text{II}} + \frac{P_{\text{II}}}{\rho}\right)$$

Since by the original assumption there are no losses due to friction, etc., between section I and II, the energy at the two sections must be equal, or

$$W\left(\frac{V^2_{\text{I}}}{2g} + h_{\text{I}} + \frac{P_{\text{I}}}{\rho}\right) = W\left(\frac{V^2_{\text{II}}}{2g} + h_{\text{II}} + \frac{P_{\text{II}}}{\rho}\right)$$

This equality agrees with Bernoulli's theory of the conservation of energy of the stream. He found, however, that his equation would not hold when ordinary liquids, such as water, were used, the right hand side of the equation always being less than the left hand side. The reason for this, he attributed to friction and other losses. Quoting him:

"Which weakens from the first, the principle of the conservation of energy. The constant equality between the increase of energy and the decrease of energy does not seem to hold. From this a hindrance can be noted which is due to friction or any form of obstacles hindering the natural flow."

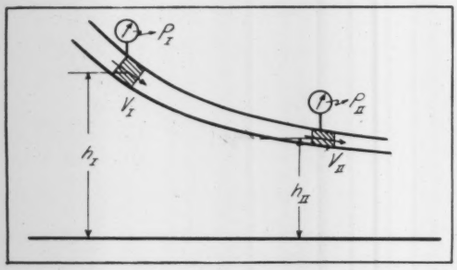

Fig. 2—Diagram for example in text which clarifies laws of fluid motion

Evaluating the energy losses and applying them to the equation, it should read

$$W\left(\frac{V^2_{\text{I}}}{2g} + h_{\text{I}} + \frac{P_{\text{I}}}{\rho}\right) = W\left(\frac{V^2_{\text{II}}}{2g} + h_{\text{II}} + \frac{P_{\text{II}}}{\rho}\right) + \text{losses}$$

This expression obeys the law of the conservation of energy and is similar to that which is quoted in many texts on hydraulics. A very useful application of the evaluation of these "losses" is in fluid flow work. Suppose water is flowing in a pipe similar to that in Fig. 2. The same two sections will be considered and the total energy at the two equated to each other. Due to friction and other losses the energy at the two sections will not be equal. This difference or loss can be found experimentally by measurement. It may be expressed either in terms of energy as Bernoulli did, or in terms of pressure.

From what has been quoted or summarized, two things are clear concerning Bernoulli's Theorem: (1) He derived it in terms of energy at two successive sections, rather than in terms of head. For a frictionless, incompressible fluid it may be stated as "the total energy at any section is equal to the total energy at any subsequent section," and (2) after trying to prove it experimentally he discovered that friction had an effect and he inserted a factor in the equation to take it into consideration. So his theorem, as first derived, is correct provided all losses are properly included, that is, the conservation of energy theory holds true for hydraulic flow of liquids.

It is also worth noting, since the work was performed almost 200 years ago, that Bernoulli assumed a pipe or vessel of any size, or shape, and placed in any position. That is, he made deductions that would fit any case.

[1] It should be remembered that all the deductions in this portion of Bernoulli's work assumes an incompressible, frictionless, and homogeneous fluid and that flow begins from rest.

Bernoulli or Bernouilli (*bĕr′ nōō yē′*) is the name of a Swiss family of scientists and mathematicians, who made their home in Basel. Daniel Bernoulli, the son of Johann (Jean) Bernoulli, was born January 29, 1700, at Gröningen, Germany, and died March 17, 1782. At an early age he studied mathematics with his father and later studied medicine. At the age of twenty-five he went to St. Petersburg as a professor of mathematics. After this he occupied a chair of anatomy and physics at Gröningen, and later chairs of anatomy, botany, and physics at Basel. Much of his time was spent with experimental and speculative philosophy. He was a member of several academies and took numerous prizes in Paris, which gives more evidence of his versatility.

Bernoulli added considerably to the principle of the parallelogram of forces which Newton and Varignon had already set forth. The assumption of the parallelism of strata (stream-line fluid flow) is accredited to him. Probably one of his most important achievements is his distinction between hydrostatic and hydrodynamic pressure. Several of his investigations are in the earlier volumes of the Commentaries Academy Petropolis. His separately published works are: *Dissertatio Inauger. Phys. Med. de Respiratione* (Basel, 1721); *Pastiones Anatomico-Botanical* (Basel, 1721); *Exercitationes Quaedum Mathematical* (Venetiis, 1724); *Hydrodynamical* (Argrentarati, 1738).

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FEBRUARY *1933*

Material in this Section prepared by Publication Committee: F. D. Mensing, *Chairman;* E. O. Eastwood and F. E. Giesecke A. V. Hutchinson, Manager of Publications

Table of Contents

Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo..... 115
Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich..... 127
Local Chapter Reports..... 132
Candidates for Membership..... 136
Candidates Elected..... 136

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Study of Summer Cooling in the Research Residence at the University of Illinois

By **A. P. Kratz**[a] **and S. Konzo**[b] ***(MEMBERS)***

Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory and the National Warm Air Heating Association, and is presented in collaboration with the paper entitled, Summer Cooling Operating Results in a Detroit Residence, by J. H. Walker and G. B. Helmrich.

The results presented in this paper were obtained in connection with the summer cooling investigation (1932) at the Research Residence[c] *(Fig. 1) at the University of Illinois, conducted by the Engineering Experiment Station of which Dean M. S. Ketchum is the director, under the supervision of A. C. Willard, Professor of Heating and Ventilation and head of the Department of Mechanical Engineering. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is due to M. K. Fahnestock, Special Research Assistant Professor, W. S. Harris, Special Research Assistant, E. L. Broderick, Research Assistant, and A. F. Hubbard, Research Graduate Assistant, for services rendered in connection with the investigation.*

THE principal objects of this investigation were (1) the determination of the cooling load and its hourly variation when cooling the Research Residence as a whole, under both night and day conditions; (2) the allocation to the various rooms of the heat entering the Residence, and the determination of the hourly variation in the cooling load of the individual rooms; and (3) the determination of the effectiveness of awnings as a means of reducing the cooling load of the Residence as a whole.

Description of the Research Residence and Cooling Plant

The Research Residence, shown equipped with awnings in Fig. 1, faces to the south and is of standard frame construction, with the exception that the studding is 2 in. x 6 in. instead of the 2 in. x 4 in. more commonly used, and that copper shingles are used on the roof. The wall section consists of weather boarding, building paper, studding, wood lath and plaster with rough sand finish. The coefficient of heat transmission for this wall section is 0.262 Btu per square foot per hour per degree Fahrenheit, at a wind velocity of 15 mph. The walls are not insulated, and no weather-stripping is used at the windows and doors. The total heated space in the winter, exclusive of the sun room, is approximately 16,390 cu ft, and the heat loss at an outdoor-indoor temperature difference of 70 F is approximately 141,000 Btu per hour including basement loss.

For the summer cooling project, the third story was closed off by means of a door at the top of the stairs, and the sun-parlor by means of a door between it and the dining room. The total space cooled, therefore, consisted of three rooms on the first story, and three on the second story, together with the two interconnecting halls, making in all approximately 14,170 cu ft. The total calculated heat gain, exclusive of sun effect, for an outdoor-indoor temperature difference of 20 F, corresponding to the most severe day for the summer, was

Fig. 1—View of Research Residence in Urbana, Ill.

[a] Research Professor, Engineering Experiment Station, University of Illinois, Urbana, Ill.

[b] Special Research Associate, Engineering Experiment Station, University of Illinois, Urbana, Ill.

[c] The Research Residence in Urbana, Ill., was built, furnished and completely equipped specifically for research work in warm air heating, by the *National Warm Air Heating Association* in December, 1924.

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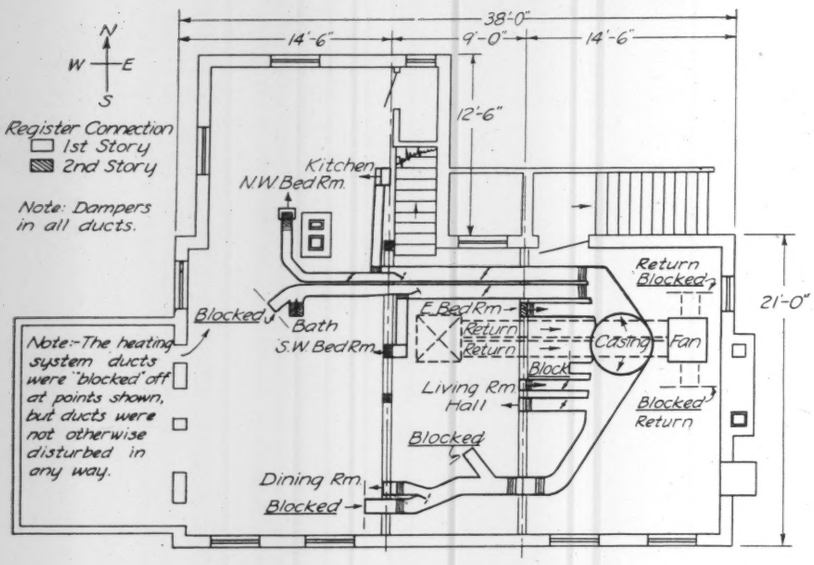

Fig. 2—Basement plan showing duct layout, for forced air system, in use for distributing cool air.

33,500 Btu per hour. The kitchen was not used for cooking and no heat was present from this source.

All of the air used for cooling was recirculated, and the mechanical warm-air system used for heating the Residence in the winter, shown in plan in Fig. 2, was utilized as the distributing system to deliver the cooled air to the rooms, and to return the warmed air to the cooling coils and fan. The latter delivered 1,475 cu ft of air per minute. With one exception, the registers on the delivery side were of the baseboard type. In the east bed room a wall type register was located 7 ft above the floor.

Part of the return air was by-passed through the cooling coils, as shown in Fig. 3, and was mixed with the balance of the return air on the suction side of the fan. The mixture was then delivered through the furnace casing into the distributing duct system. Control was obtained by means of a modulating by-pass damper operated from a room thermostat. Thus the temperature of the air at the registers was regulated to maintain the required dry-bulb temperature in the rooms.

For the purpose of these tests, a plant using ice as the cooling agent was selected in preference to one using mechanical refrigeration, because it was considered that the weight of ice melted could be conveniently and accurately determined, while the determination of the refrigeration load for the mechanical plant offered complications in testing that might be reflected in the accuracy of the results. No consideration was given to any possible relative merits in the two types of plants from commercial or other standpoints. The water circulated through the cooling coils was, therefore, cooled by means of ice.

Two sections of cooling coils, each section consisting of three rows of finned tubing set in headers, were used. The coils contained a total of 154 linear feet of ¾ in. tubing having helical fins, spaced 6 fins per inch. The overall diameter, including fins, was 1½ in. The inside dimensions of the casing were 27⅞ in. by 49⅛ in. Since the two coils were used in tandem, the gross or *face* area presented to the flow of air was 9.5 sq ft, and the net free area was 4.2 sq ft. The total cooling surface was 473 sq ft. A drip pan at the bottom of the casing for the coils, as shown in Fig. 3, served to collect the condensation resulting from the dehumidification of the air. From the drip pan, the condensation was drained into a bucket placed on scales.

The arrangement of the cooling plant is shown in Fig. 4, and the details of the ice-tank in Fig. 5. The water was pumped from the bottom of the ice-tank and passed through the cooling coils. The latter were connected in series, and the flow of the water was in the opposite direction to that of the air, thus forming a counter-flow arrangement. After passing through the coils, the water was returned to spray heads in the top of the ice-tank, and sprayed over the ice. The latter

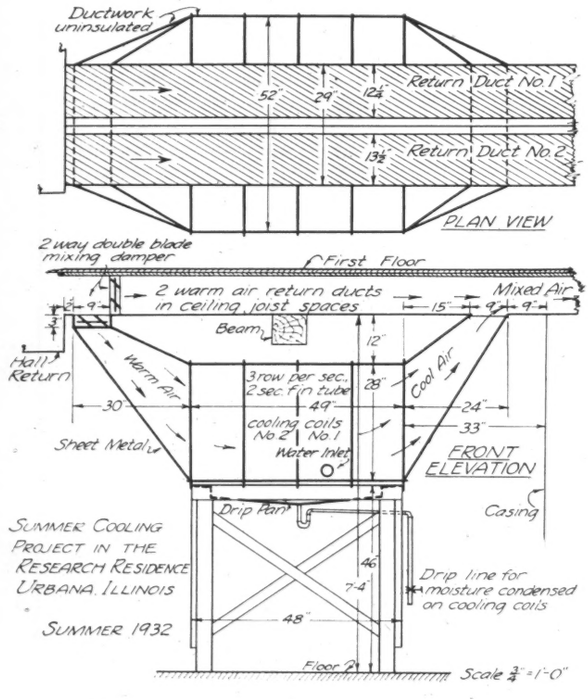

Fig. 3—Details of cooling coils in main unit.

was placed on racks, and both the circulating water and the meltage accumulated in the bottom of the tank. This water space was connected to a weir box, shown as Section *B-B* in Fig. 5, containing a drain pipe, which faced upward with a sharp edge that served as a weir and maintained the water at a constant level just below the ice racks. All of the water that drained out of the weir box, therefore, represented the ice meltage that occurred during the time that the drain was open. By allowing this water to collect in a pan on scales, it was possible to determine the ice meltage for any period of time desired.

For the purpose of these tests, the Research Residence was equipped with three temperature recorders which gave continuous records of the air temperature outdoors, at three points on the first story, at two points on the second story and at one point on the third story. In addition, a recording psychrometer gave a continuous record of the relative humidity on the first story.

For the work in the individual rooms, two commercial portable room coolers were used. These were placed on scales reading to 0.02 lb in order to determine the ice meltage and water resulting from dehumidification.

For part of the tests, the Research Residence was equipped with a total of 19 awnings, which were placed on the front, or south exposure, and on the two ends, or east and west exposures. The awnings were of the hood type with rope pull-up, consisting of a top and two wings with an 11 in. valance all around. They were made from 10 oz duck, painted on one side only with gray and green stripes.

Method of Conducting Tests

For a few of the preliminary tests, an attempt was made to maintain the schedule of desirable indoor temperatures corresponding to outdoor temperatures given as Table 2, page 10, in the A. S. H. V. E. GUIDE 1932. It at once became evident, however, that after the outdoor temperature had reached a peak value and started to decline, it was necessary to progressively reduce the indoor temperature in order to maintain the schedule. This procedure imposed a load on the plant that was regarded as outside of the practical limits of operation. Furthermore, if the conditions indoors were conducive to comfort during the peak outdoor temperature, there was nothing to indicate that the maintenance of these conditions resulted in discomfort after the peak had been passed.

The following method of operation was therefore adopted as a standard. The cooling plant was started when the effective temperature indoors rose to 75 deg, irrespective of the outdoor temperature. This occurred

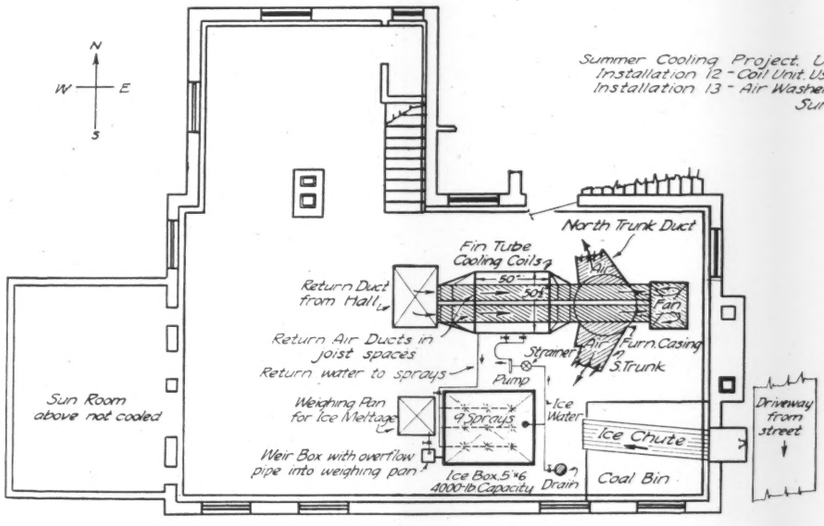

Fig. 4—Line diagram of ice tank, cooling coil unit, and return air ducts in the basement of the Research Residence, Urbana, Ill.

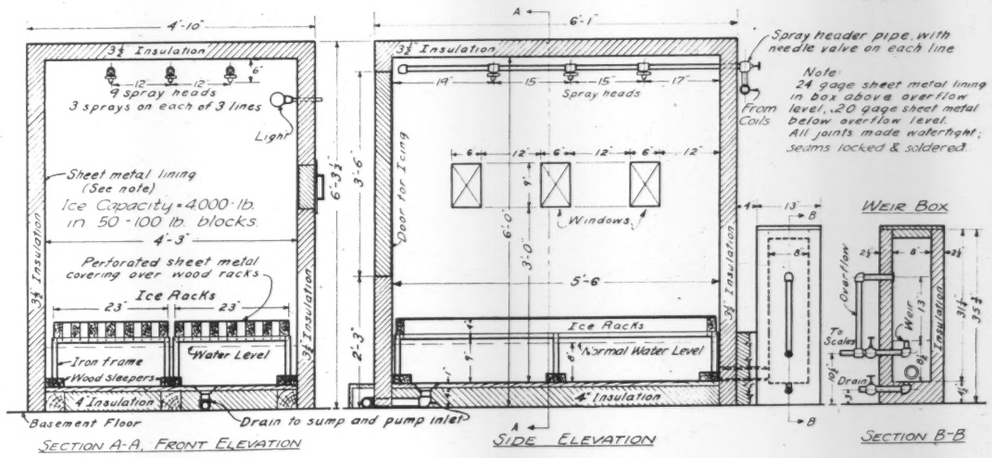

Fig. 5—Details of ice tank and weir box, showing arrangement of ice racks, water spray pipes and piping.

when the indoor dry-bulb temperature reached a value between 78 and 82 F, depending on the relative humidity. After starting the plant, the dry-bulb temperature was maintained approximately constant until the outdoor temperature dropped to about the same value. The plant was then stopped. Owing to the reduction in relative humidity after the plant had been in operation for about 1½ hours, the resulting effective temperature corresponding to the operating dry-bulb temperature of from 78 to 82 F was approximately 72 deg.

A calibration test on the plant, with the by-pass damper opened to pass the maximum amount of air through the cooling coils, indicated that if the temperature of the air at the registers was not allowed to fall below 60 F, 396 gal of water per hour had to be circulated through the coils. Since the plant was controlled with the by-pass damper and it was not possible, within the limits of the time available, to determine the optimum amount of water to be circulated under various conditions, all tests were run with 396 gallons of water circulated per hour at a pressure of 23 lb per square inch. The water entered the coils at about 35 F.

The dampers in the ducts were set to balance the plant with all of the room doors closed. When the plant was operated with the room doors open, the balance was not disturbed.

During the whole summer from June 1 to October 1 either continuous records or periodic readings were made of the temperature of the outdoor air, the temperature of the air in the basement and on each story, the temperature of the surface of the roof, the surface of the ceiling in one second story room, the relative humidity indoors and outdoors, and the times of starting and operating the plant. These readings were plotted on a continuous chart with time as a base, and to a scale of 0.4 in. per hour. This chart furnished a complete history of the variations in indoor and outdoor conditions, both during the periods of tests and the periods preceding and following tests.

For the heat load allocation tests, the unit room coolers were located in the rooms to be studied and these rooms were closed off from the rest of the Residence. The coolers were operated on the on-and-off principle to maintain the desired temperature in the rooms, and the central cooling plant was operated to maintain the same temperature in the rest of the rooms and halls. On account of the small amount of cooling weather obtained during the summer, the amount of data obtained with the unit coolers was entirely too meagre to establish any conclusion in regard to the allocation of the heat to the various rooms, or the hourly variation in the cooling load to these rooms.

General Results

In general, the results indicated that the ducts of a central forced-air heating system could be successfully adapted as a distributing system for cooled air without material alteration. Some readjustment of dampers was necessary in order to balance the plant when the cooled air was circulated. It was necessary to adjust the dampers so that more air was delivered to the second story than to the first. When the plant was balanced, the rooms were uniformly cooled with no indication of the existence of a pool of cold air near the floor, and as the sun changed position, the balance was not materially disturbed when the plant was controlled with a single thermostat placed in the dining room. An average temperature difference of approximately 3.5 deg was observed between the floor and the ceiling, and 1.8 deg between the floor and the breathing level, with the cooler air near the floor. The air was delivered from the registers at temperatures varying from 60 to 70 F, depending on the outdoor temperature, and at average velocities varying from 50 to 450 fpm measured with an anemometer placed one inch from the register face. No objectionable drafts were observed in the rooms, except in one case in which the air was delivered from a baseboard register at a velocity of approximately 450 fpm. Even in this case, conditions were easily corrected by employing a baffle in front of the register face in order to direct the air flow toward the ceiling. The best distribution of cooled air in the room was obtained with the wall register placed 7 ft above the floor. In this case, the velocity of the air leaving the register was approximately 350 fpm. The gain in temperature of the air, from the fan inlet to the register faces did not exceed 3 deg, and condensation did not appear at any time on the furnace casing or basement duct system. No insulation was used on these surfaces. The basement temperature remained practically constant at 70 F.

The cooling season at Urbana, Illinois, during the summer of 1932, extended from June 1 to October 1, with a total of 62 days on which the maximum temperature reached 85 F or more. This represented a total of 1,471 degree-hours above 85 F. A total of 43.3 tons of ice was used for the season. The maximum hourly rate of ice meltage observed was 220 lb. This rate occurred on Test P-29, when the Residence was not equipped with awnings, and for which the maximum outdoor temperature attained was 95 F. The plant had been in operation for the previous 24 hours. The maximum meltage for any 24-hour period was 4,243 lb. This occurred on test P-11, when the Residence was equipped with awnings, and for which the maximum outdoor temperature attained was 103 F. In this case, however, the plant had not been in operation for the 24 hours previous to the test.

Since the total volumes of air and water circulated were not varied for the different tests, the electrical input to the fan and pump motors remained practically constant. The power consumed by the fan motor was ¼ kw, and by the pump motor was 1¼ kw. The power required by the pump was influenced by the pressure of 23 lb per square inch required to operate the spray heads in the ice tank. The use of these spray heads was found necessary in order to break the water up into a sufficiently fine spray to prevent the formation of craters in the ice. These craters filled with water and interfered with the accuracy in the determination of the ice meltage. The gain in accuracy resulting from the elimination of the craters, however, was obtained by a sacrifice in the operating efficiency of the plant as a whole, due to the pressure required by the spray heads. In a service plant, where the accurate determination of the ice meltage is not necessary, the spray heads could be dispensed with and the water showered over the ice in larger streams, thus reducing the power for the pump.

Table 1—Complete Summary of Tests

Date	Test No.	Start of Test	Length of Test Hours	Outdoor Temperature			Indoor Temp Aver.	Aver Temp Diff In—Out	Aver. Humidity		Ice Meltage		Dehumidification Total Lb	Hourly Heat Leads					
				Max During Test	Min During Day	Aver During Test			Outdoor Aver During Test	Aver Indoors	Equiv Total Lb	Corr for Basement Loss Lb		Uncorrected		Corr. for Basement		Corr. for Basement	
														Total	Sensible	Total	Sensible	Total	Sensible
														Btu per Hour				Btu per Degree per Hour	
7-7	P-4	10:45A	8.0	88.0	73.5	83.8	77.7	6.1	46.2	63.4	1060	701	32.19	19225	15000	12625	8400	2070	1377
7-9	P-5	1:15P	10.0	87.0	62.5	81.2	75.5	5.7	69.7	56.1	1459	1001	38.24	21000	16980	14400	10380	2526	1821
7-10	P-6	1:25P	4.0	90.5	70.5	86.3	77.2	9.1	65.6	60.4	472	289	20.51	16975	11590	10375	4990	1140	548
7-11	P-7	2:00P	6.0	85.0	66.5	81.5	78.2	3.3	47.4	53.3	907	632	27.82	22885	18010	16285	11410	4937	3456
7-12	P-8	2:00P	7.5	93.5	59.5	86.4	77.0	9.4	58.1	51.9	1318	975	41.40	25300	19500	18700	12900	1989	1372
7-13	P-9	9:30A	13.5	99.0	70.0	91.0	78.4	12.6	58.0	50.8	2642	2024	97.96	28180	20560	21580	13960	1712	1108
7-14	P-10	11:40A	14.25	99.0	75.0	89.4	81.1	8.3	62.8	49.2	3050	2397	113.23	30820	22470	24220	15870	2917	1912
7-15	P-11	10:00A	24.0	103.0	78.0	88.8	80.8	8.0	59.9	46.2	4243	3144	111.56	25450	20400	18850	13800	2356	1725
7-16	P-12	10:00A	24.0	97.0	77.5	83.2	80.0	3.2	59.1	45.1	2713	1614	76.48	16280	12930	9680	6330	3026	1978
7-17	P-13	10:00A	12.0	90.5	72.5	85.8	80.7	5.1	37.3	44.0	1261	711	30.31	15130	12480	8530	5880	1673	1153
7-20	P-14	9:00P	24.0	98.0	78.0	86.7	80.0	6.7	60.5	42.9	3530	2431	109.04	21170	16410	14570	9810	2174	1464
7-21	P-15	9:00P	24.0	97.5	74.0	83.9	79.2	4.7	55.4	43.3	3320	2221	88.94	19920	16030	13320	9430	2833	2005
8-10	P-20	9:30A	12.0	91.0	67.0	83.1	76.4	6.7	73.4	48.8	1761	1211	61.11	21120	15775	14520	9175	2166	1369
8-15	P-21	9:30A	10.5	87.0	62.0	81.8	75.6	6.2	71.5	50.4	1238	757	35.49	16975	13425	10375	6825	1674	1101
8-16	P-22	9:00A	13.5	90.0	72.0	84.0	77.1	6.9	76.3	48.6	2025	1407	60.71	21600	16880	15000	10280	2173	1489
8-23	P-24	10:00A	11.0	87.5	57.5	80.9	74.9	6.0	52.5	44.4	1838	1334	40.64	24060	20180	17460	13580	2908	2262
8-24	P-25	10:00A	11.0	90.0	60.0	83.5	75.1	8.4	52.3	46.0	1464	960	35.01	19170	15830	12570	9230	1496	1098
8-25	P-26	9:00A	13.0	89.5	69.0	83.6	77.3	6.3	63.5	48.1	1578	983	45.01	17470	13835	10870	7235	1725	1148
8-26	P-27	9:00A	12.0	87.0	67.5	80.8	76.6	4.2	69.0	46.5	1513	963	41.44	18155	14525	11555	7925	2751	1886
8-29	P-28	7:00A	24.0	94.0	75.0	82.7	78.0	4.7	74.2	47.2	3680	2581	104.77	22090	17500	15490	10900	3295	2318
8-30	P-29	7:00A	24.0	95.0	73.0	81.9	77.9	4.0	72.2	47.4	3752	2653	109.39	22500	17720	15900	11120	3975	2780
8-31	P-30	7:00A	24.0	93.0	71.5	79.0	77.1	1.9	74.1	45.9	3237	2138	87.15	19420	15600	12820	9000	6750	4737

Notes: Basement loss approx 6600 Btu per hour or equal to an ice meltage of 45.8 lb per hour. Total Correction = (No. of hours) (45.8).
Test P-16 on 7-22-32 and Test P-23 on 8-17-32 omitted due to unfavorable change in weather conditions.
Test P-17, P-18, and P-19 on 7-26, 7-28, and 8-5-32 respectively omitted on account of incomplete data.

The cooling load in the basement consisted of the heat gain of the coils, furnace casing and duct system, and the electrical load imposed by the fan and pump motors and the lights. Separate determinations of this load indicated that it remained practically constant at 45.8 lb of ice per hour, or 6,600 Btu per hour.

The principal results for all tests are given in Table 1. It has not been possible to use all of these for the curves and analyses. The lengths of tests were necessarily determined by the character of the weather. Furthermore, the total ice meltage in the case of short tests was largely influenced by the history of the Residence previous to the start of the test. On some of the shorter tests, the starting load formed an excessive proportion of the total ice meltage for the test. Hence, many inconsistencies may be observed in comparisons based on the averages and totals for individual tests. It was therefore necessary, for the purpose of analysis, to group the tests made under comparable conditions, and in some cases the analysis has been based on the maximum load extending over a 4-hour period including the maximum outdoor temperature. Some comparisons have been based on the load corrected for basement losses, since this represented the part of the load influenced by the particular factor under consideration. Furthermore, this basement load would vary for different plants and, by especial attention to features of design favoring higher operation efficiency rather than adaptability to accurate measurement of test data, could be very materially reduced.

In the test plant, the basement load was augmented by the necessity for the use of spray heads in the ice tank and by the use of more basement lights than would be demanded by a service plant. Also, during the test , an effective temperature of approximately 72 deg was maintained in the Residence. This condition was perfectly comfortable with the occupants clothed in light summer clothing. No member of the staff experienced any ill effect either from shock due to sudden changes in environment or from contracting a cold. However, the indications were that somewhat less cooling would be satisfactory for residence service. Just how much less is debatable at present. Hence, it is possible that by varying the design and operation from that used for the test plant, a reduction in ice meltage could be made. The figures presented, however, may be considered as representative, with a reasonable margin of safety, of the cooling requirements for a structure similar to the Research Residence.

A complete graphic log covering a period of two 24-hour tests with the house equipped with awnings is shown in Fig. 6, and a similar graphic log for two 24-hour tests with the awnings removed is shown in Fig. 7. Fig. 6 contains all of the data from the corresponding portion of the continuous chart previously mentioned, with the addition of effective temperatures, ice meltage

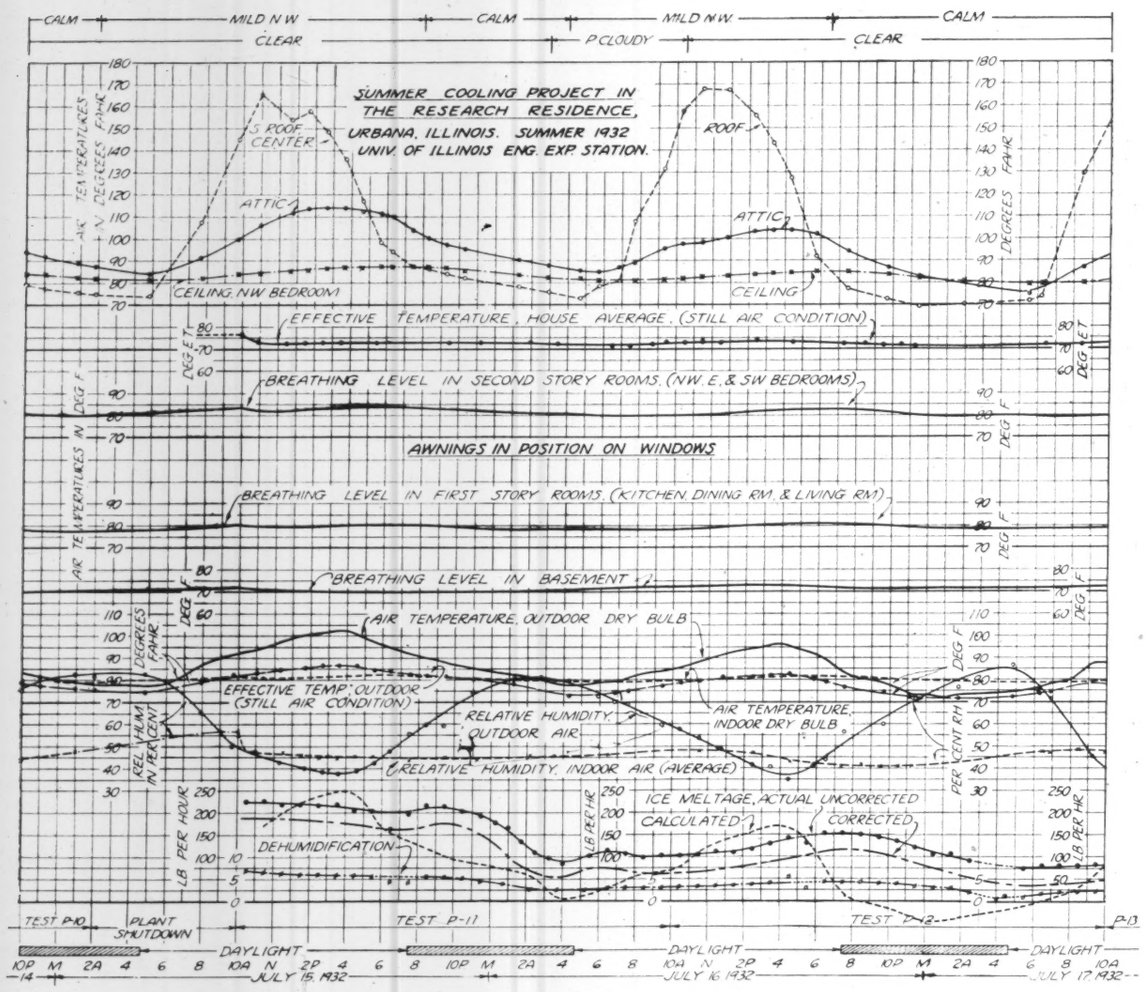

Fig. 6—Graphic log of observations—July 15, 16 and 17, 1932.

and calculated cooling load. In Fig. 7, the data on roof, attic, and ceiling temperatures have been omitted.

From Fig. 6 it is interesting to note that the temperature of the copper shingles on the roof attained a value as high as 169 F. At times values as high as 175 F were observed. The temperature of the air in the attic space rose to 112 F. The time shown between the attainment of maximum temperature for the roof surface and that of maximum air temperature in the attic does not represent the time lag of the attic space, however. The data for the roof were taken from the south slope, while the attic space was in a north wing, and the west slope of the roof above it did not receive the sun until afternoon. The time at which the maximum was attained in the attic space corresponded approximately with that for the maximum outdoor temperature, thus indicating comparatively no lag. The temperature of the surface of the ceiling below the attic reached a maximum about 3 hours later than that of the attic itself. In this case, the ceiling was insulated with one inch of insulating quilt, and the surface temperature never rose more than approximately 3 deg above the temperature of the air in the room below it. The fact that the ceiling below an attic space becomes in effect a large panel radiator, thus adding to the discomfort of the occupants of the room independently of the air temperature, indicates a necessity for effective insulation outside of the desirable reduction in cooling load accompanying its use.

Relative Humidities and Dehumidification

Fig. 6 illustrates the effect of the indoor relative humidity on the starting load on the plant. The plant was not operated on the day previous to the start of test P-11. The indoor relative humidity rose gradually until it reached a value of 57 per cent at 10 a. m., July 15, at which time the outdoor temperature was 92 F and the indoor temperature was 81 F, corresponding to an indoor effective temperature of 76 deg. At this time the cooling plant was started for test P-11. At the end of an hour's operation, the relative humidity had dropped to about 46.5 per cent, and at the end of four hours had stabilized at about 45 per cent. Meanwhile the indoor effective temperature dropped from 76 deg to 72 deg while the indoor dry-bulb temperature remained comparatively constant at about 81 F. The dehumidification, appearing as condensation on the coils, decreased from about 7 lb per hour to 5.5 lb per hour. Data on other tests showed decreases from as high as 9 lb per hour to as low as 4.5 lb per hour within two hours after the

start of the test. The average dehumidification load ranged from approximately 3,000 to 8,000 Btu per hour, and formed a considerable part of the total load. It averaged about 4,000 Btu per hour, and constituted approximately one-fifth of the total load. A number of calculations, based on the condensation weighed after the indoor relative humidity had stabilized, and at outdoor temperatures near the peak, gave results for the number of air changes ranging from 0.7 to 1.0 air changes per hour. These results were subject to changes in moisture content of the walls, floors, and furniture, but the consistency of the weighed amounts, and the fact that long periods of uniform indoor relative humidities were maintained, indicates that three-quarters of an air change per hour represents the amount of air infiltration with a fair degree of accuracy.

Comparison of Actual and Calculated Cooling Loads

The variation in the actual cooling load, as determined by ice meltage from hour to hour, may be observed from Figs. 6 and 7. From these curves it may be noted that while the outdoor air temperature reached a maximum at about 4 p.m., the actual load reached a maximum from two to four hours later. This was particularly true if the plant had been operated during the 24 hours previous to the start of a test. The effect of the sun on the load may be observed by comparing the shape of the load curves in Figs. 6 and 7. In the case of Test P-12, shown in Fig. 6, for which the Residence was equipped with awnings, the load increased gradually to a peak occurring four hours later than the time for the maximum temperature for the day. In the case of Tests P-28 and P-29, shown in Fig. 7, the load rose suddenly, reaching one peak at about noon, then declined somewhat and increased to reach a second peak at 5:30 p.m. The first peak was probably caused by the sun, which showed its maximum effect at 10 a.m., and the second peak by a combination of the sun, which reached a secondary maximum at 2 p.m., and the heat lag of the structure. The total heat radiated through the windows, based on the glass area and exposure of the windows and on the curves given by F. C. Houghten[1], and others was as follows: 6 a.m., 980; 8 a.m., 11,587; 10 a.m., 15,040; 12 m., 11,290; 2 p.m., 14,065; 4 p.m., 10,093; 6 p.m., 828 Btu per hour. A limited number of observations on the solar radiation, a sample of which is shown in Fig. 8, indicated that the solar radiation at Urbana, Illinois, which is in practically the same latitude as Pittsburgh, Pennsylvania, agreed reasonably well with the results obtained at Pittsburgh by the authors previously cited[1]. Their curves were therefore accepted as applying at Urbana. The observations of solar radiation were made with an Abbot silver disk pyrheliometer,* a description of which is given by J. H. Walker[2] and others.

A comparison of the actual cooling load, based on measured ice meltage, and the calculated cooling load reduced to terms of ice meltage, may be made from Figs.

* Loaned by the Smithsonian Institution, Washington, D. C.
[1], [2] See Bibliography.

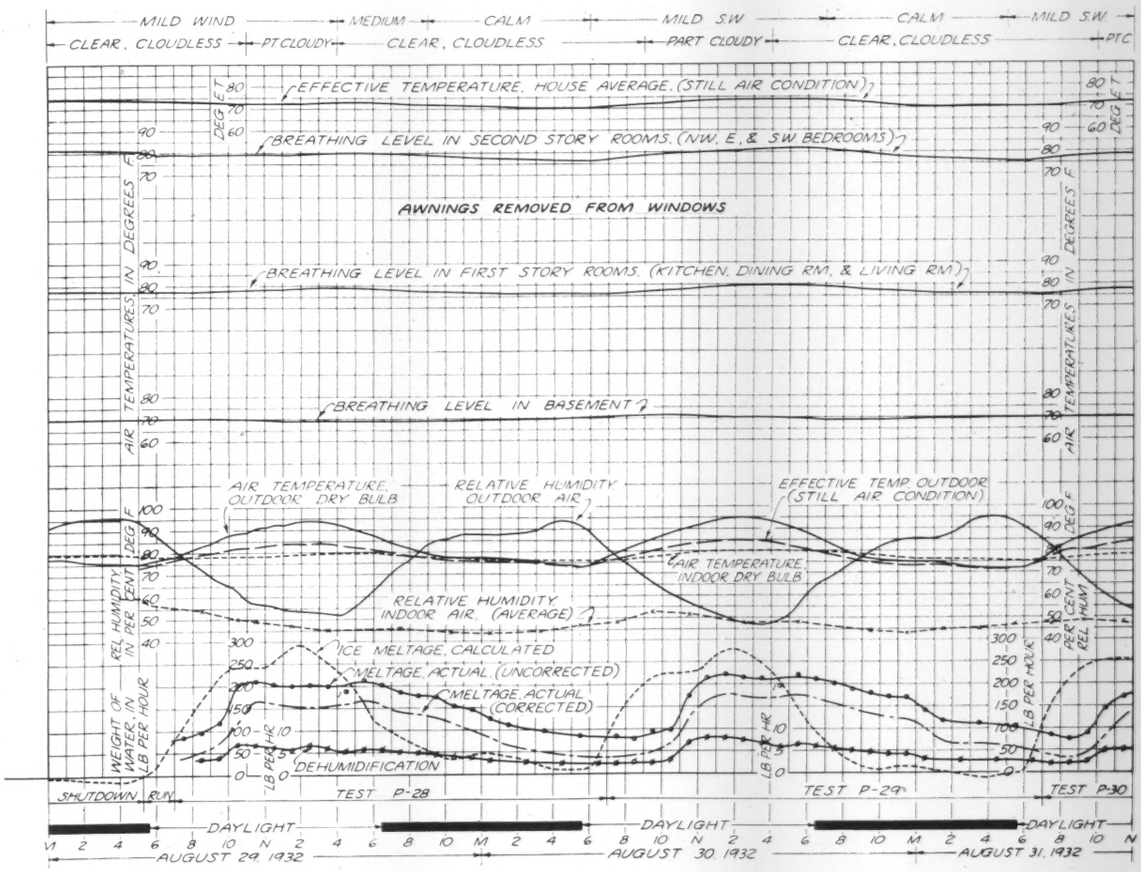

Fig. 7—Graphic log of observations—August 29, 30 and 31, 1932.

6 and 7. This comparison should be based on the cooling load corrected for loss in the basement, since the calculated load applies only to the portion of the Residence above the basement. The calculated load at any given time consisted of the sum of the sun load, the dehumidification load, the load due to occupancy, the load due to sensible heat from air infiltration, and the heat transmission load. The latter was based on the actual outdoor-indoor temperature difference shown by the curves at that time and the heat transmission coefficients applying to the walls, windows, floors and ceilings of the Residence. The sun load was based on the glass exposures for the Residence and on the solar radiation

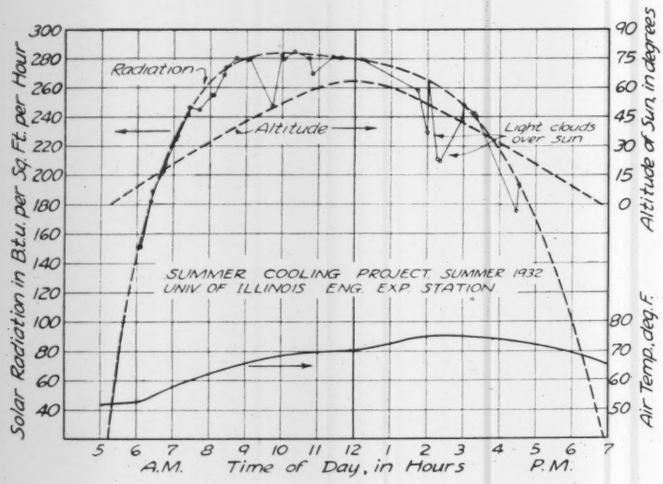

Fig. 8—Solar radiation and outdoor air temperature on a clear, bright day. August 19, 1932, Urbana, Ill.

obtained from the curves of Houghten[1] and others corresponding to the given time, with the assumption that a single thickness of glass intercepted 17 per cent of the total radiation. The latter was determined experimentally and checked closely with the figure given by Walker[2] and others. In the case of the Residence equipped with awnings, the latter were considered as completely intercepting the sun's radiation, thus eliminating the sun load caused by radiation through the windows. The dehumidification load and the load due to sensible heat in air infiltration were based on the existing indoor and outdoor air temperatures and relative humidities, and on the assumption of three-quarters of an air change per hour for the 14,170 cu ft of space cooled. Occupancy was assumed to be between two and three people, or 1,600 Btu per hour.

From these curves, it is evident that the heat lag of the structure operates in such a way as to make it almost futile to attempt to calculate the cooling load for the building at any specified time. The actual load never attained a value as high as the calculated maximum load corresponding to the maximum outdoor temperature. Furthermore, the true load at the time that the maximum load actually occurred, was much greater than the calculated load corresponding to this time. The curves all showed one point in common, however. That is, the calculated load at approximately 6 p.m. agreed with the actual load at this time, and corresponded to a value only slightly less than the actual maximum load. It is therefore possible that the maximum load for the purpose of design, could be based on the sun effect and the probable outdoor temperature at 6 p.m. for any given locality. Whether or not this is a feasible method of calculation, and whether it would apply to structures differing in character from the Research Residence is admittedly somewhat problematical.

The fact that the area between the curves in excess of the actual load is approximately equal to the area representing the deficiency over actual load, suggests the possibility of obtaining an average load by calculation, based on the average temperature differences, relative humidities and sun effect spread over 24 hours. This average sun effect, consisting of the total Btu during the hours of sunshine divided by 24, was 5,300 Btu per hour, or 35 per cent of the maximum. For Test P-29, the calculated average cooling load, based on 24 hours, was equivalent to 116 lb per hour of ice meltage. The actual average ice meltage was 111 lb per hour. Hence, the calculated meltage was 4.5 per cent in excess of the actual. This method has no merit as a means of determining the maximum load, since the maximum hourly meltage for this test was 220 lb per hour. However, if the seeming incongruity of considering the sun load as distributed uniformly over 24 hours is disregarded, the method might have a possible application in estimating the probable daily meltage.

Effect of Outdoor Temperature on Cooling Load

An attempt to correlate the cooling load with the average outdoor-indoor temperature differences shown in Table 1 resulted in many inconsistencies, due to the fact that this average temperature difference was influenced by the character of the daily temperature curve, and the relation between the maximum outdoor temperature and balance of the outdoor temperatures. Hence it was possible to obtain approximately the same average temperature with radically different maximum temperatures. Also the total or average cooling load was dependent on the heat lag and previous history of the structure, and on the portion of the day over which it was possible to run the test. Therefore, a number of tests were selected in which the start of the test occurred a sufficient length of time previous to that for attainment of maximum outdoor temperature to permit conditions to become stabilized before the maximum outdoor temperature was reached. For this group of tests, the cooling load per degree difference in temperature was based on the average outdoor-indoor temperature difference for the four hours included between two hours before and two hours after the time for the maximum, or peak temperature. These results are tabulated in Fig. 9 and have been plotted against the average temperature over the four hour peak period in Fig. 10.

From Fig. 10 it is evident that the cooling load is not directly proportional to the difference in temperature, but for the same difference in temperature the cooling load increases as the outdoor temperature increases. This was true both for the sensible and the total cooling load, the latter including dehumidification, and whether or not the house was equipped with awnings. This is

TEST NO	DATE 1932	LENGTH OF TEST	WEATHER SKY	SUN	WIND	AVER TEMP OVER 4 HOUR PEAK PERIOD Outdoor Air	Indoor Air	B.T.U. PER HR PER DEG DIFF. IN OUTDOOR-INDOOR TEMP. UNCORRECTED FOR BASEMENT TOTAL	SENSIBLE	CORRECTED FOR BASEMENT TOTAL	SENSIBLE	RATIOS OF COOLING LOAD S	T	PERCENT REDUC. IN LOAD WITH AWNINGS S	T
SERIES A. AWNINGS IN POSITION. CLEAR BRIGHT DAY															
9	7-13	13.5	Clear, few cum	Bright	Mild	97.5	78.8	1650	1192	1297	840				
11	7-15	24.0	Clear	"	"	101.3	81.9	1660	1330	1320	990				
12	7-16	24.0	"	"	"	94.8	81.8	1627	1282	1120	775				
14	7-20	24.0	Few Cum.	"	"	95.5	80.7	1646	1254	1199	808				
					Average			1646	1265	1234	853				
SERIES B. AWNINGS REMOVED. HAZY DAY WITH SCATTERED GREY NIMBUS CLOUDS															
21	8-15	10.5	Cldy	Dull	Calm	84.9	76.7	2273	1779	1468	975				
26	8-25	13.0	"	"	Med	87.5	77.9	2135	1661	1448	974				
					Average			2204	1720	1458	975				
SERIES C. AWNINGS REMOVED. BRIGHT DAY WITH SCATTERED WHITE CUMULUS CLOUDS															
22	8-16	13.5	Part Cldy with few white cumulus clouds	Bright but sun in and out	Calm	89.3	78.0	2107	1638	1524	1055				
27	8-26	12.0			Med	85.8	76.9	2205	1742	1464	1000				
28	8-29	24.0			Mild	93.1	79.0	2063	1608	1596	1140				
29	8-30	24.0			Med	94.2	79.8	2167	1612	1708	1153				
30	8-31	24.0			Med	91.9	78.5	2004	1582	1512	1088				
					Average			2109	1636	1561	1087				

AVER. CORRECTED FOR BASEMENT 0 400 800 1200 1600

SERIES A: T 1234, S 853 — 0.78 | 0.79 | 22 | 21

SERIES B: T 1458, S 975 — 0.90 | 0.93 | 10 | 7

SERIES C: T 1561, S 1087 — 1.00 | 1.00 | 0 | 0

NOTES: Awnings on all East, South, and West windows. 19 in number. Windows closed and locked; window shades drawn half way down.
Total cooling load is the sum of the "dehumidification load" and the "sensible heat load".
Basement correction for heat load includes heat losses in basement from ice tank, coils, piping, etc.
Values in above table were based on results obtained during the 4-hr. period of max. outdoor temperature.
All 4-hr. peak periods were preceded by several hours of actual plant operation.

Fig. 9—Effect of awnings on reduction of cooling load.

equivalent to a statement that the cooling load per degree temperature difference is not constant, and that any method of calculation based on Btu per degree-hour and seasonal degree-hours is incorrect unless some account is taken of the increase in cooling load per degree difference in temperature as the actual outdoor temperature increases.

Effect of Awnings

Fig. 9 shows the reduction in cooling load effected by the use of awnings on all east, south, and west exposures. This comparison has been made between the cooling loads corrected for basement loss, since the only portion of the load affected by the awnings was that above the basement, and the basement load would vary with different individual plants. Comparing the averages for the two groups of tests run on bright, practically cloudless days, it may be noted that the awnings effected a reduction in sensible cooling load of 22 per cent and in total cooling load of 21 per cent. It may be noted that the tests without awnings, P-21 to P-30 inclusive, were run nearly one month later than the ones with awnings. A calculation of the total amount of heat from the sun

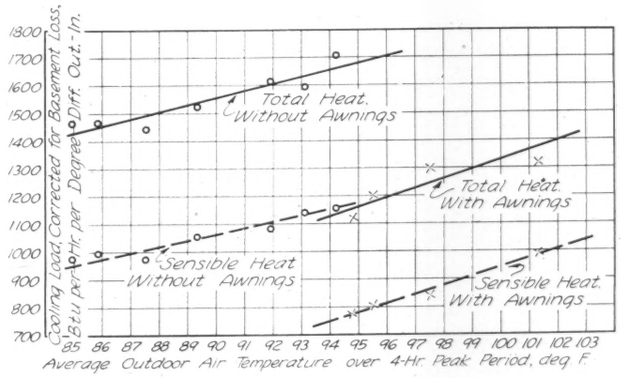

Fig. 10—Variation in cooling load with outdoor air temperature during 4-hour peak period.

radiated through the windows, based on the curves of Houghten[1] and others, indicates that the heat from this source is somewhat greater during September than during August. Since the awnings intercept all of the sun load imposed through the windows, it is immaterial whether these tests were run during August or during September. However, the load obtained without awnings during September was probably somewhat greater than it would have been if the tests had been run during August. The comparison that has been made, is equivalent to assuming that both series of tests were run during September. If it had been possible to compare two series of tests made in August, the load with awnings would have been the same, but the load without awnings would have been somewhat less than the one actually used. To this extent therefore the comparison as made has tended to favor the awnings. To offset this, however, the average outdoor temperature was about 6.5 deg higher during August, for the group of tests with awnings than during September for the one without awnings, and since the cooling load per degree difference in temperature increases with the outdoor temperature, the comparison as made has tended to be less favorable to awnings. This is illustrated in Fig. 10, from which it may be noted that the reduction in cooling load resulting from the use of awnings is approximately 30 per cent for outdoor temperatures between 93 and 97 per cent. Hence, it is evident that the actual percentage of saving effected by awnings is dependent on what part of the season is selected for comparison, but it is probable that the figures based on the comparison as made are conservative for the average reduction of load resulting from their use. It is possible that the results may be affected by the variations in the sun effect on the walls from month to month, but it is not possible to separate this for the purpose of analysis.

Referring to Figs. 6 and 7, it may also be noted that the awnings changed the character of the daily cooling load curve, and reduced the extent of the period of

Table 2—Summary of Total Days, Hours and Degree-Hours Above Base Temperatures of 85, 90 and 95 F for Summer of 1932 at Urbana, Illinois

Month	Total Days			Total Hours			Total Degree-hours		
	85 F and Above	90 F and Above	95 F and Above	85 F and Above	90 F and Above	95 F and Above	Above 85 F	Above 90 F	Above 95 F
June	18	5	.....	72.0	16.6	.....	222.8	30.5	.....
July	22	14	8	163.8	81.7	29.7	929.9	327.8	72.9
August	18	6	.....	90.6	23.5	.....	316.3	50.8	.....
Sept.	4	.....	.....	2.7	.....	.....	1.7	.....	.....
Total	62	25	8	329.1	121.8	29.7	1470.7	409.1	72.9

maximum cooling load from a duration of approximately 6 hours to one of two hours.

Seasonal Cooling Load

Table 2 gives a summary of the number of days, hours, and degree-hours above bases of 85, 90 and 95 F for the summer of 1932 at Urbana, Illinois, based on data observed at the Research Residence. From this it is evident that the choice of a base temperature is an important item in determining the number of degree-hours in a cooling season. If it is assumed that no cooling will be required until the outdoor temperature reaches 90 F then the summer of 1932 would have had a total of 409.1 degree-hours requiring cooling. If, however, a base of 85 F is assumed, then the summer would have had a total of 1470.7 degree-hours, or 3.6 times the amount for a 90 F base.

The uniformity of the curves of outdoor temperature plotted against the time of day as in Figs. 6 and 7 suggested the possibility that there might be some correlation between the maximum daily temperatures and the number of hours and degree-hours above any given base temperature. The daily outdoor temperatures, in general, followed a cycle similar to the one shown as an inset in Fig. 11. Therefore, if a given base temperature was selected, the length of the chord between the two points at which this base temperature line intersected the curve of outdoor temperatures, represented the number of hours for the given day that temperatures at, or above, the base temperature persisted. Also, the area between the base line and the temperature curve above the base line, shown cross-hatched in Fig. 11, represented the number of degree-hours above the given base. This procedure was followed for each of the 62 days during the summer of 1932 for which the maximum temperature reached 85 F or above, using three separate base lines, at 85, 90 and 95 F. The resulting number of hours and degree-hours for each day were then plotted against the maximum outdoor temperature for that day, as shown in the two groups of curves in Fig. 11. It was then possible to draw three curves, for base temperatures of 85, 90 and 95 F, representing the average of the points with a fair degree of accuracy. The points accordingly represent the deviations of the actual cycles for the individual days from the ideal average cycle represented by the curves. As an example, if a base temperature of 85 F is selected, and the maximum temperature of 92 F was attained on a given day, then from the upper group of curves in Fig. 11, a total of 7.2 hours with temperatures above 85 F could be expected for that day, and from the lower group of curves, a total of 30 degree-hours would be probable. There is no reason to believe that the character of the daily temperature cycles used in constructing these curves is peculiar to Urbana, Illinois, alone, and therefore it is probable that the curves in Fig. 11 are applicable to a wide range of localities. Hence, it should be possible to estimate the probable number of hours and degree-hours above a selected base for any season in a given locality, by counting the total number of days occurring at various maximum temperatures from the U. S. Weather Bureau reports for the locality, and multiplying the number of days thus obtained for each maximum temperature by the number of hours or degree-hours obtained from the curves in Fig. 11 corresponding to the various maximum temperatures selected. The summation of the hours or degree-hours thus obtained represents the total for the season.

An estimation of the number of degree-hours above 85 F for each of the past ten summers at Urbana, Illinois, has been made in this manner, and is shown in Table 3. In compiling this table, the various ranges of outdoor temperatures have been grouped as shown in Column 1. The mean temperatures for the various ranges are given in Column 2, and the number of degree-hours per day, as read from the curves in Fig. 11 corresponding to the mean temperatures in Column 2, are

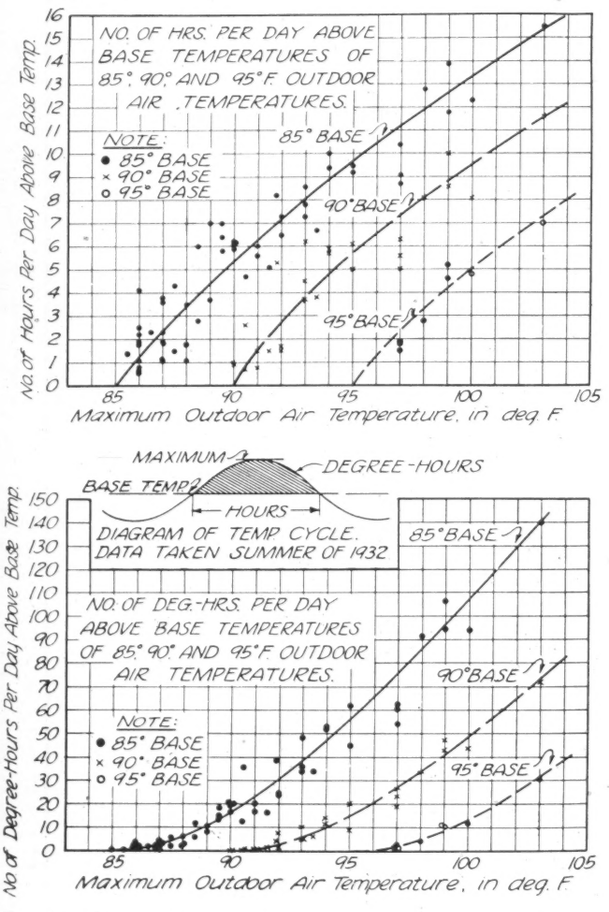
NO. OF HRS. PER DAY ABOVE BASE TEMPERATURES OF 85°, 90° AND 95° F. OUTDOOR AIR TEMPERATURES.
NOTE:
• 85° BASE
× 90° BASE
○ 95° BASE
85° BASE
90° BASE
95° BASE
No. of Hours Per Day Above Base Temp.
Maximum Outdoor Air Temperature, in deg. F.
MAXIMUM
BASE TEMP.
DEGREE-HOURS
HOURS
DIAGRAM OF TEMP CYCLE. DATA TAKEN SUMMER OF 1932
NO. OF DEG.-HRS. PER DAY ABOVE BASE TEMPERATURES OF 85°, 90° AND 95° F. OUTDOOR AIR TEMPERATURES.
No. of Degree-Hours Per Day Above Base Temp.
Maximum Outdoor Air Temperature, in deg. F.

Fig. 11—Curves showing relation between maximum outdoor air temperature and number of degree hours and number of hours per day above base temperature.

given in Column 3. In the two columns for each year shown, the first column is the total number of days included in the months of June, July, August, and September, having maximum temperatures falling between the limits shown in Column 1, and the second column is the product of the number of days and the corresponding number of degree-hours per day from Column 3. The first three columns in this table may be considered as applicable to any locality, while the balance of the table applies to Urbana only. The ranges could be made smaller than those shown in Column 1, but greater accuracy at this point is hardly warranted when it is considered that the final result is only a reasonably close approximation at best.

That it is a reasonably close approximation is indicated by the fact that the actual number of degree-hours for the summer of 1932, calculated from the summation of the degree-hours for the individual days was 1,471, while the number calculated from the average curves and shown in Table 3 was 1,538, or a difference of 4.2 per cent. A similar approximation for the number of hours above 85 F gave 318 as compared with the actual number of 329, or a difference of 3 per cent.

From Table 3 it is evident that the cooling season is extremely variable, even in the same locality. During the ten years, it varied from a maximum of 2,502 degree-hours to a minimum of 327 degree-hours. The latter represents only about one-eighth of the maximum. Furthermore, during this ten-year period there were four years in which the number of degree-hours was less than one-fourth of the maximum, six years in which it was less than one-third and eight years in which it was less than two-thirds of the maximum. That the cycle represented by the ten years selected presents nothing unusual in the way of variation is shown by Fig. 12, in which the average of all of the maximum temperatures for each day in the months of June, July, August, and September are plotted against the years, over a period of 45 years. It may be noted also from Fig. 12, that the shape of the degree-hour curve, plotted from the date in Table 3, closely approximates the shape of the average maximum temperature curve. Hence, the variations shown by the 10-year cycle selected may be regarded as typical of the variations that might occur in any 10-year cycle during the whole period of 45 years, and both the curves and the table serve to emphasize the futility of any attempt to calculate the probable cooling load for a given season from a mean value of the number of degree-hours based on the average for a cycle of seasons.

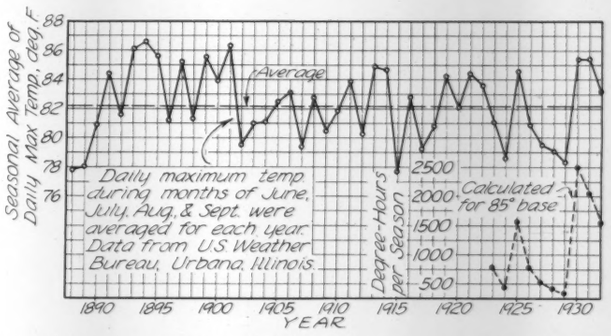

Fig. 12—Graphical record of summer temperatures in Urbana, Illinois, during years 1888 to 1932.

Daily Variation in Outdoor Temperatures

It may be noted from the outdoor temperature curves shown in Figs. 6 and 7, that the difference in temperature between the maximum in the daytime and the minimum during the preceding night ranged from 19 to 28 deg, and that for a period of six hours, or more, the outdoor temperature was at least 20 deg below the maximum attained during the same day. An analysis of all of the daily temperatures during the months from June 1 to October 1 proved that, with very few exceptions, a difference of at least 20 deg was exhibited. The exceptions occurred when the maximum was 85 deg or less. The average difference when the maximum was from 75 to 85 F was 19 deg. For maximums between 85 and 95 F, the average difference was 23 deg, and for maximums above 95 F it was 25 deg. This indicated that for considerable periods at night the outdoor air constituted a reservoir from which air could be drawn in order to cool the residence at night, and reduce the cooling load required during the following day. For a large portion of the time, enough cooling could probably be effected in this way to make it unnecessary to start the cooling plant the next day, particularly in the case of an insulated structure. It should be emphasized, however, that in order for this method to be effective, a fan large enough to circulate sufficient outdoor air to cool

Table 3—Number of Days of Stated Maximum Temperature and Number of Degree Hours above 85 F for Season from June 1 to October 1 at Urbana, Illinois

Range of Maximum Outdoor Temperature Deg F	Mean of Range	No. of Deg Hrs Above 85 F Per Day (From Fig. 11 at Mean of Range)	1932*		1931		1930		1929		1928		1927		1926		1925		1924		1923	
			No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours	No. of Days	Total Deg Hours
85– 87	86	1.0	23	23	22	22	15	15	17	17	19	19	12	12	14	14	18	18	12	12	14	14
88– 90	89	10.5	16	168	14	147	13	137	15	158	14	147	10	105	17	179	20	210	12	126	12	126
91– 93	92	30.5	11	336	9	274	15	457	5	152	8	244	6	183	8	244	17	518	4	122	7	213
94– 96	95	56.0	4	224	18	1008	7	392	0	0	0	0	4	224	6	336	8	448	3	168	6	336
97– 99	98	86.0	6	516	7	602	7	602	0	0	0	0	0	0	0	0	3	258	0	0	1	86
100–102	101	119.0	1	119	0	0	5	595	0	0	0	0	0	0	0	0	1	119	0	0	0	0
103–105	104	152.0	1	152	0	0	2	304	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total			62	1538	70	2053	64	2502	37	327	41	410	32	524	45	773	67	1571	31	428	40	775

* Data for 1932 based on results at Research Residence.
Data for 1923-1931 from U. S. Weather Bureau.

the walls and contents of the structure, and not merely to cool the air inside of the building, would be required. Just what could be accomplished in this respect was not determined by this investigation, but the results indicate that this would be a fruitful field for future investigation.

Results of Tests on Unit Coolers

The data obtained from the unit coolers were too meager to permit any conclusions in regard to allocation of the cooling load. Some data were obtained on the operation of these units, however, and may prove of interest. On continuous operation, one unit cooled three rooms to a dry-bulb temperature of 76 F when the outdoor temperature was 85 F. The relative humidity at this time was 55 per cent outdoors and 59 per cent indoors. The ice meltage was 36.1 lb per hour, and the air passing through the unit was cooled from a dry-bulb temperature of 76 F to one of 65 F. The dehumidification amounted to about 0.25 lb per hour and the wet-bulb temperature of the entering air was 66 F and of the leaving air was 61 F, or a reduction of 5 deg in the wet-bulb temperature.

On intermittent operation, this unit cooled one room to a dry-bulb temperature of 75 F when the outdoor dry-bulb temperature was 81 F. Both outdoor and indoor relative humidities were 55 per cent. The overall ice meltage was 20.1 lb per hour. The meltage on the standby periods was 6.1 lb per hour and during the actual running periods was 31.5 lb per hour. The time of actual running was 44.5 per cent of the total time. In this case, the unit reduced the dry-bulb temperature of the air passing through it from 75 F to 60.5 F. The wet-bulb temperature of the air passing through the unit was reduced from 63.5 F to 55.2 F, or 8.3 deg. The dehumidification was approximately 0.25 lb per hour.

Conclusions

The following conclusions may be drawn, subject to the limitations of the conditions under which the tests were run:

1. The ducts and registers of a central forced-air heating system can be successfully adapted as a distributing system for cooled air in the summertime without material alterations.
2. There is no tendency for a pool of cold air to collect near the floor if the temperature of the air entering the room through the registers is not lower than 60 F.
3. Baseboard registers can be used without objectionable drafts resulting if the velocity of the air at the register face is below 300 fpm although wall registers located 7 ft above the floor are preferable to baseboard registers.
4. A building of the type of the Research Residence may require the equivalent of two tons of ice in 24 hours including the basement load on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.
5. An effective temperature somewhat higher than 72 deg may be satisfactory for residence service, in which case more conservative results for the cooling load can be obtained.
6. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.
7. The cooling load per degree difference in temperature is not constant, but increases as the outdoor temperature increases.
8. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.
9. The seasonal cooling requirements are extremely variable from year to year, and the ratio of the degree-hours exhibited by any two seasons occurring within a ten-year period may be as high as 7.5 to 1. Hence an average value for the degree-hours per season is comparatively meaningless.
10. The results of these tests suggest two phases, in regard to which further investigation is desirable. The first concerns the load requirements when unit coolers are used in one or more partially isolated rooms with the rest of the building not cooled. The second concerns the use of a fan at night either to provide more comfortable conditions during the following day without provision for cooling, or to reduce the load required for cooling during the following day.

Bibliography

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2. A. S. H. V. E. Research paper entitled, Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford and E. P. Wells. (*Heating, Piping and Air Conditioning,* February, 1932, pp. 125-137.)

Death of Hosford D. Kellogg

Word has been received of the death of Hosford Dudley Kellogg, a Life Member of the Society and a widely known figure in the heating industry in Philadelphia, at his home in Haverford, Pa.

Mr. Kellogg was born at Granville, Mass., and at the time of his death was 73 years of age, having been in ill health for the last six years. For more than half a century he was associated with the H. B. Smith Co. and was manager of the Philadelphia Office of that company for 40 years until September 1, 1927, when he retired from active business.

He joined the Society in 1916 and was granted Life Membership in 1929. Mr. Kellogg is survived by his widow, a daughter, Imogene, and a son, Hosford D., Jr., to whom the Officers and Council express their sorrow for the loss they have sustained.

Death of James F. Sprague

In the passing of James F. Sprague in Milwaukee, Wis., on December 2, 1932, at the age of 72, the Society lost one of its valued members—a man who was known as the dean of heating engineers in that section of the country.

Mr. Sprague had a long and useful career in his chosen profession. He was born June 14, 1860, at Boston, Mass., and was graduated in 1878 from the Boston Latin School, where he took a mechanical engineering course. For six years he studied architecture under Prof. F. M. Clark in Boston and in 1884 became superintendent for Shepley, Rutan & Coolridge in Boston and St. Louis. He served as assistant architect on the St. Louis Board of Education from 1888 to 1896, when he became associated with B. F. Sturtevant Co. in Chicago. He remained in Chicago for 17 years, being later identified with the Automatic Heating Co. and the Chicago Health Department. In 1913 he transferred his activities to Milwaukee with the Bayley Blower Co. and in 1927 formed the Sprague Engineering Co., of which he was president. He has been responsible for many important heating, ventilating and air conditioning installations in the west and, although his affiliation with the Society was comparatively short, he had many friends and acquaintances among its members.

The Officers and Council wish to extend their sincere sympathy to his widow, daughter and son, who survive.

Summer Cooling Operating Results in a Detroit Residence

By J. H. Walker[1] (*MEMBER*) and G. B. Helmrich[2] (*NON-MEMBER*)
Detroit, Mich.

This paper is presented in collaboration with the paper entitled, Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo

THIS paper gives the results obtained from the study of a residence cooling system under normal conditions of occupancy and with only a moderate degree of cooling as compared with the conditions maintained in the Research Residence at the University of Illinois.[3] It was particularly desired to determine the minimum requirements for producing a moderate improvement in comfort and to determine the feasibility of cooling an entire residence with a cooling system requiring only a small investment. The work was undertaken by The Detroit Edison Company.

There is some difference of opinion as to the extent to which artificial cooling of residences is economically feasible. Some believe that the cooling of a first floor living room and one bedroom is all that should be attempted, because of the high cost of installing a complete system or because of the operating cost. It is true perhaps that if it is necessary to install a system of pipes or air ducts in an existing house the installation cost may restrict the number of rooms which may be cooled. In a house which has a warm air heating system, however, the existing ducts offer an excellent means of distributing the cooled air, and thus there is a large number of homes in which the application of summer cooling is a relatively simple problem. The operating cost is reasonable if only a moderate amount of cooling is done, at least in a normal summer in the vicinity of Detroit, Michigan; and a moderate amount of cooling throughout the entire house is considered preferable to concentrating the effect in two or three rooms.

It is obvious that both the economics of residence cooling and the design of cooling systems will vary widely in different parts of the country and general conclusions should not be drawn from the results in this one installation. The results should be indicative, however, of what may be expected from installations of the same type, operating under similar conditions.

The Residence

The residence in which the tests were made is located in Birmingham, a suburb of Detroit. It is of moderate size and of substantial frame construction. The heating system is a mechanical warm air system having return ducts from all of the rooms.

The cooling was done by means of ice which was chosen because it was believed that the ice cooling system is appropriate for this kind of service. A system using ice is simple and inexpensive to install. The cost of ice is rather high compared to the cost of operating an individual refrigerating machine, but the operating season is of short duration. Ice also possesses the merit of flexibility as to the rate of refrigeration. By the simple expedient of varying the melting rate the cooling effect can be changed through a wide range.

The only additional equipment which was necessary to provide for complete cooling in this residence was a tank for the ice, a coil placed in the air circuit, and a pump to circulate the ice water through the coil. Actually other features were added for experimental purposes but the tank, coil, and pump are the only essential elements, and the experience gained indicates that all of these could have been smaller and more simply arranged. The actual installation cost was comparatively high be-

Fig. 1—The Detroit residence is a typical well-built, frame house of moderate size

Note: Dimensions 46 ft by 26 ft. Eight rooms cooled—4 on each floor. Total space cooled 17,200 cu ft. All openings weatherstripped. Walls built up of shingle siding, 3 ply waterproof building paper, pine sheathing, studding, lath and plaster, with flexible insulation nailed between studding. Attic floored over with flexible insulation between attic floor joists. Shutters or awnings on all windows exposed to sun. Winter heat loss 93,000 Btu per hour for 70 F inside and 0 deg F outside

[1] Superintendent of Central Heating, The Detroit Edison Company.
[2] Engineer, Construction Bureau, The Detroit Edison Company.
[3] See Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo.
Presented at the 39th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

cause of the necessity of providing a new fan and rearranging the ducts, but it is estimated that, had the cooling equipment been installed at the time the house was built, the added cost would not have exceeded $500.00, and might have been as low as $300.00.

The arrangement of the equipment is shown in Fig. 2. The ice tank is buried, without insulation, just outside the basement wall. It is charged through a trap door

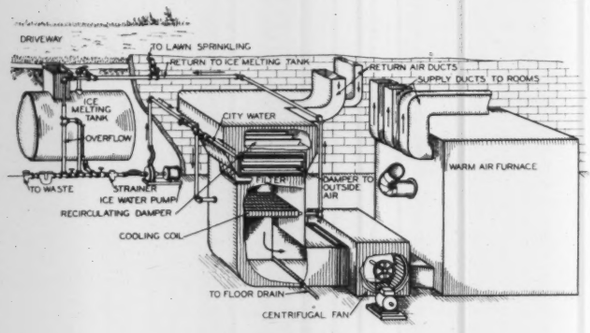


Fig. 2—The cooling apparatus

Notes: Furnace Fan—Centrifugal type, rated at 2,600 cfm at ⅜ in. static pressure. Outlet velocity 1,250 fpm at a speed of 482 rpm. Motor ½ hp capacitor type.
Ice Tank—5 ft 7 in. long by 4 ft diameter.
Cooling Coils—Standard automobile radiator having 492 sq ft of surface.
Pump—Centrifugal pump rated at 10 gpm against a total head of 35 ft. Driven by ½ hp capacitor type motor.
Note that the required sizes and capacities of these units in most cases proved to be considerably smaller than actually installed. (See text.)

which is close to the driveway and therefore readily accessible to the delivery truck. The cooling coil is a standard automobile radiator. The pump is an inexpensive centrifugal pump, mounted on rubber discs and connected with the piping by rubber hose. The fan is a standard design of double inlet centrifugal furnace fan and is set on strips of soft rubber, while the driving motor is set on cork to insure quietness of operation.

Provision for Outside Air

At all times when artificial cooling was done the system was operated with complete recirculation, no outdoor air being used. Provision was made, however, to draw in outside air when desired and this feature proved to be extremely valuable, and should be considered an essential part of such a system. In Detroit, as in most parts of the country, there are many nights and a few days during the summer when the outside air is cooler than the inside air, but there is no breeze to bring it into the rooms. The connection for bringing in outside air was simple to provide and satisfied all cooling requirements at certain times for merely the slight cost of the fan power.

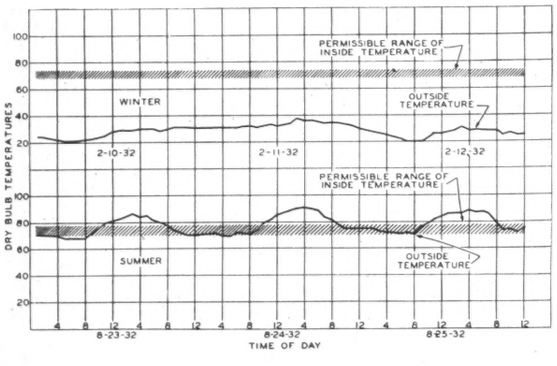
Cooling Is a Cyclic Process

Engineers are inclined to regard cooling problems as the reverse of, but similar to, heating problems and to consider the process of cooling as that of maintaining the interior of a building at a constant temperature and absorbing heat which flows inward much as heat flows outward in winter. There are several essential differences between the heating and cooling processes however. One difference is that there is a wide range of temperatures through which people are reasonably comfortable in summer, whereas in winter they are accustomed to having indoor temperatures controlled within rather close limits.

Another difference concerns a point already mentioned, that in a large part of the country the conditions which exist out-of-doors during many days, and especially during many nights, in summer are entirely comfortable and all that is required is to create those conditions indoors by drawing in outside air; no artificial cooling is required. Thus while heating is required continuously in winter, summer cooling is necessarily intermittent and to a great extent cyclic. This is well illustrated by Fig. 3 which shows the relation between outdoor temperatures and acceptable indoor temperatures in summer and in winter. There are of course occasional periods in Detroit and elsewhere when the night temperatures are high; but there were none in Detroit during the past summer and only a very few in the hot summer of 1931.

Temperature Rise Without Cooling

In view of the cyclic nature of the cooling process and the well-known lag in the temperature rise in a tight, well-shaded house, the question naturally arose as to what the indoor temperature would be if no artificial cooling were done. This house was well protected from the sun, being equipped with either awnings or shutters on all windows exposed to direct sunlight; and the walls were well insulated. How effective these provisions are is well illustrated in Fig. 4. Although the outside temperature during this test was not extremely high the slow rise of the inside temperature is quite evident. The



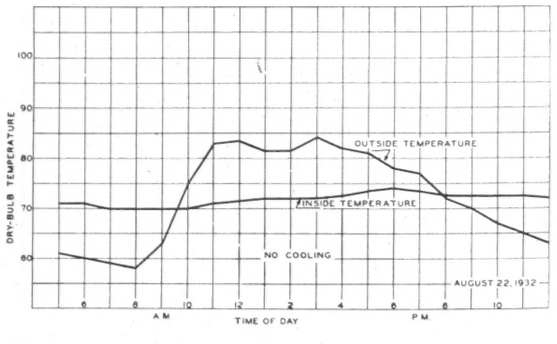


Fig. 3 (above)—Relation of outdoor temperatures to desirable indoor temperatures for winter and summer conditions

Fig. 4 (below)—Inside temperature when no cooling was done

average rate of rise of the first floor temperature was only about ½ deg per hour for a differential between inside and outside of approximately 10 deg.

Window shading, insulation, and thermal capacity of the structure are only partially effective in keeping the house cool, however, and could not be considered to obviate the need for artificial cooling. Eventually, during a protracted hot spell the inside temperature would become uncomfortably high.

Method of Operation of the Cooling System

The fan was operated experimentally at various speeds but at the most satisfactory point it delivered approximately 2,110 cfm, equivalent to an average air change of over seven times per hour for the entire space cooled. The temperature of the air leaving the registers was from 59 to 68 F, averaging about 65 F during the cooling period. The velocity varied from 100 to 317 fpm.

The temperature of the water entering the coil was 47 F which was low enough to cause a considerable amount of moisture to be condensed from the air. The indoor humidity ranged from 45 to 60 per cent while the system was in operation which seemed to be entirely satisfactory, there being no noticeable feeling of clamminess at any time. The dehumidifying load in a residence is of course normally small compared to the cooling load except for occasional temporary loads caused by cooking and bathing.

The air supply registers were all located just above the baseboard. There were no seriously uncomfortable drafts except where the arrangement of the furniture was such that the occupants found themselves in the immediate vicinity of the registers. It would have been desirable to have had deflectors in front of a few registers but balancing the system by adjusting dampers quite effectively moderated objectionable drafts. There seemed to be no serious temperature stratification of the air in any of the rooms and no noticeable short-circuiting of the air to the outlet grilles. The temperature difference between the floor level and the breathing line, measured on different occasions, averaged about 4 deg.

Results Obtained

During the early part of the summer the system was operated freely and without particular regard to economy and the ice consumption, on days when the outside temperature reached 95 F, varied from 1,000 to 2,500 lb. Indoor temperatures of 75 F to 77 F were held continuously throughout the day and while these conditions seem ideal to a continuous occupant, to a person entering or leaving there was a greater temperature differential than was entirely comfortable. The cost of operation with ice at $5.00 per ton with this unrestricted cooling was of course rather high.

The results obtained on two typical days with this type of operation are shown in Fig. 5. The ice consumption was 1,000 lb on one day and 2,500 lb on the other day with inside temperatures maintained at not above 78 F and 77 F respectively.

It soon became apparent that there was a very wide range of temperatures that could be maintained in the house which could be considered as satisfactory, and it

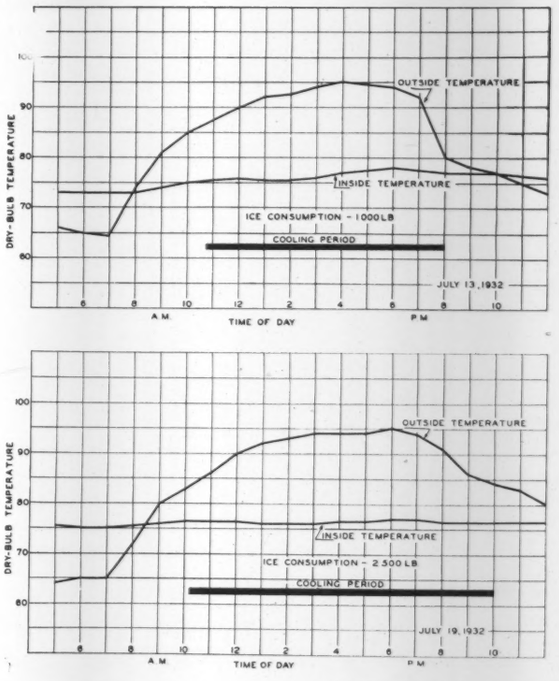


Fig. 5—Results with unrestricted use of the cooling system

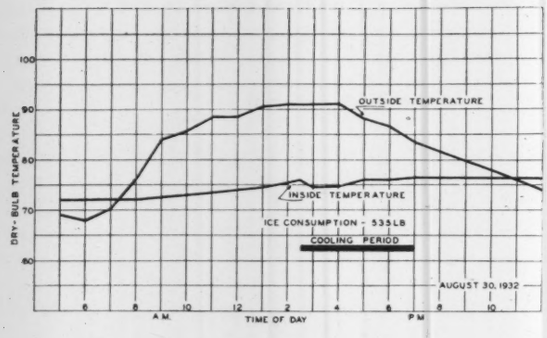
was further evident that to maintain temperatures as low as 76 F and 78 F on the hottest days required an ice consumption that was out of all proportion to the slightly increased comfort obtained as compared with a slightly higher temperature. Therefore during the month of August a more moderate amount of cooling was done and the ice consumption was thereby greatly reduced.

The rate of ice consumption was limited by covering part of the coil surface with a sheet metal baffle and by throttling the flow of water through the coil. The ice was not delivered each day until afternoon and then in a limited quantity of 500 to 700 lb. Thus a moderate cooling effect was obtained. The temperature of the house rose gradually as the outside temperature rose, but when the cooling system was put in operation by starting the pump, the rise of the house temperature was checked. This method tended to give what is probably the best temperature regulation, that is, an increasing differential between inside and outside temperature as the outside temperature increases.

One feature was noted in these tests that is usually true—it is impracticable to attempt to hold a large *differential* between outdoor and indoor temperatures when the outdoor temperature is falling. It would be practically impossible to bring down the inside temperature at a rate commensurate with a rapidly falling outside temperature.

With the type of system described, the proper method of operation is to stop the flow of water through the coil when the outside temperature is falling in the late afternoon but to continue to circulate the air with the fan. When the outside effective temperature becomes less than that inside, the outside-air damper should be opened and outside air drawn in.

The results obtained with this moderate amount of cooling are shown in Figs. 6 and 7. They are highly important as illustrating what can be done with a reason-



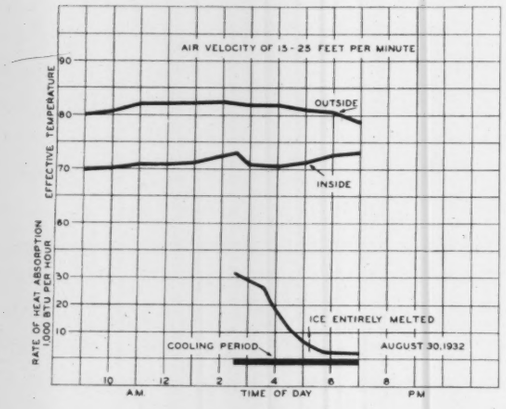


Fig. 6 (above)—Inside and outside dry-bulb temperature for August 30, 1932

Note that a satisfactory condition was obtained on this day with a consumption of only 535 lb of ice

Fig. 7 (below)—Inside and outside effective temperatures for the same day illustrated in Fig. 6

able consumption of ice. The conditions were entirely satisfactory and probably much better physiologically than a lower temperature would have been. The ice consumption on this day was 535 lb and on other similar days it varied from 500 to 700 lb. These experiments thoroughly demonstrated the important fact that by the use of a moderate amount of ice a satisfactory degree of comfort could be obtained. Table 1 shows the complete operating data for the day represented in Figs. 6 and 7.

An incidental feature of the installation was a provision for using city water instead of ice as the cooling medium. City water, if its temperature is low enough, is an economical medium for cooling. In this case the water came from deep wells and had a temperature of 58 F during June, which rose to about 61 F in September. A few tests were made which indicated that on days which were not extremely hot, a considerable amount of cooling could be done in this way. The piping was so arranged that the water used for cooling could then be utilized for lawn sprinkling.

The use of city water appears to have only a limited possibility of application, however. In many localities its temperature is unsuitable and in others there would undoubtedly be restrictions placed on its use for this purpose; but where cold water is available it is a relatively economical source of cooling. Comparing, for example, the cost of operation with ice, city water, and an electrically-operated compression refrigerating machine, the

Table 1—Operating Data for a Day with Moderate Cooling, August 30, 1932

Time of cooling	2:30 p.m.—7:00 p.m.	
Pounds of ice used	535	
Ice melting time— hours	2.75	
Ice melting rate—pounds per hour	195	
Pounds of water condensed from air	12.87	
Inside Temperatures—First Floor	*For 24 Hr.*	*For Cooling Period*
Maximum	77.2 F	77.2 F
Average	74.8 F	76 F
Average wet bulb	.....	66.8 F
Outside Temperatures		
Maximum	91 F	91 F
Average	80 F	88 F
Average wet bulb	.....	74.8 F
Average temperature of air to coil	71.2 F	
Average temperature of air from coil	64.8 F	
Average temperature of air from grilles	65.2 F	
Average velocity of air from grilles—fpm	190	
Air circulated—cfm	2,110	
Air changes per hour	7.3	
Average power input to motors—kw	0.84	
Total kwh consumed	3.78	
Minimum temperature of water to coil	48.5 F	
Minimum temperature of water from coil	55.0 F	
Quantity of water circulated—gpm	4.93	

figures of Table 2 were obtained, using the prevailing local unit costs for ice, water and electricity.

Overall Results for Entire Summer

The method of operation of the system was varied considerably during the summer and the total ice consumption is, therefore, not representative of the best operating method, being somewhat higher than necessary for a moderate amount of cooling. However, the actual summer's results, as given in Table 3, are interesting and show a surprisingly reasonable operating cost.

Temperature Control

The question of temperature control naturally arose and required some experimentation. A thermostat, main-

Table 2—Calculated Comparative Operating Costs With Different Methods of Cooling

	Ice	City Water (Well Water)	Electricity (In a 2-ton Compression Unit)
Unit cost	$5.00 per ton	$0.20 per M gal	$0.0225 per kwh
Heat absorbing capacity	344,000 Btu per ton[a]	50,000 Btu per M gal Initial temp 58 F Discharge temp 64 F	10,236 Btu per kwh[b]
Cost per M Btu of cooling effect	1.45 cents	0.40 cents	0.22 cents

[a] Based on ice water temperatures rising to 60 F at end of run (all water recirculated).

[b] Equivalent to 33.5 kwh to produce the same cooling effect as a ton of melting ice—coefficient of performance of compression unit assumed to be 3.

taining a constant temperature, was obviously unsuitable because the inside temperature should be permitted to rise when the outside temperature rises. The simple method finally adopted was to adjust the system for a moderate rate of heat absorption or, in other words, a moderate ice melting rate, and to operate continuously for a few hours each day, permitting the temperature to rise as it would. Control is thus obtained by regulating the number of hours of operation per day. The system was put into operation by merely starting the fan and pump by means of conveniently located switches and was operated until the outside temperature fell in the afternoon or until the ice was exhausted. The cost of operation was controlled by putting into the ice tank each day only the amount of ice found to be sufficient for a moderate amount of cooling.

Relation of Cooling Requirements to Degree-hours

The factors which affect cooling load and ice consumption are very complex. In general the cooling and dehumidifying load has two components—the external load which is largely a function of outside temperature and the load which is due to internal sources and is nearly independent of outdoor conditions. In a residence the internal load is usually negligible.

It would be convenient to have an approximate rule for determining the ice or current consumption, similar to the degree-day method of calculating fuel consumption for heating. It was noted that the cooling system was operated, using ice, on those days when the outside temperature at some time during the day exceeded 85 F.

Table 3—Operating Data for Cooling Season, 1932

Total number of days when artificial cooling was used	22
Total number of hours when artificial cooling was used	135
Total equivalent amount of ice used*	18,166 lb
Average ice consumption per day of artificial cooling	826 lb
Average ice consumption per hour of actual operation	135 lb
Temperature, during days of artificial cooling	
Maximum inside temperature at any time	80 F
Average inside temperature	74.7 F
Maximum outside temperature	95 F
Average outside temperature for 24 hours	77.8 F
Average outside temperature for cooling periods	83.7 F
Average dry bulb temperature difference between indoors and outdoors during hours of operation	8.1 F
Average effective temperature difference between indoors and outdoors during hours of operation	6.2 F
Power Use	
Electricity used for pump and fan, entire season	94 kwh
Electricity used per ton of ice melted	10.4 kwh
Approximate amount of electricity used to manufacture the ice, at 50 kwh per ton	454 kwh
Estimated amount of electricity which would have been used in an individual refrigerating machine to produce the same cooling effect	304 kwh
Total electricity used (94+454)	548 kwh
Total estimated use of electricity, assuming individual refrigerating plant (94+304)	398
Costs	
Ice (including equivalent of city water) 9.08 tons at $5.00 per ton	$45.40
Electricity 94 kwh at 2¼ cents per kwh	$ 2.12
Total operating cost	$47.52
Average total cost per day when cooling was used	$ 2.16
Average cost per hour of actual operation	$ 0.352

*i.e. including an amount of ice equivalent to the cooling effect of the city water which was used on certain days.

If therefore 85 F is arbitrarily taken as the base temperature and the degree-hours above 85 F computed, the following comparison with the ice consumption is obtained:

	Degree-Hours Above 85 F	Ice Consumption (Lb)	Per Cent of Total Degree-Hours	Per Cent of Total Ice Consumption
July	339	8,950	67	65
August	164	4,782	33	35
Total	503	13,732	100	100

This fairly close agreement between degree-hours and ice consumption, as expressed in per cent, may be mere coincidence or may indicate that the method used is a simple yard stick for determining cooling requirements at various temperatures. It is interesting to note the figures for preceding years for degree-hours above 85 F in Detroit, as follows:

Year	Total Degree-Hours Above 85 F
1932	529
1931	1,341
1930	935
1929	490
1928	260
1927	334
1926	351
1925	521
1924	224
1923	402
Ten year average	538.7

Miscellaneous Notes on Operation

The capacity of some of the items of equipment was found to be considerably in excess of the required capacity. This was true of the pump, which was finally throttled down to a flow of 6 gpm. The ice tank could also have been smaller if daily ice deliveries are assumed. A capacity of 45 cu ft would have been ample. The coil, too, proved to be about 50 per cent larger than necessary for the ice melting rate finally used.

Experiments in which some of the air was by-passed around the cooling coil indicated that such by-passing was unnecessary. The essential purpose of a by-pass would be to permit the mixing of some relatively warm recirculated air with the air which is cooled to a low temperature for dehumidification, so that the resultant temperature of introduction into the rooms would not be uncomfortably low. It was found, however, that with a water temperature of 47 F sufficient dehumidification was obtained for comfort and the air temperature at the registers was 62 F which was not uncomfortably low.

A feature that would probably further reduce the operating cost would be to cool only the first floor in the day time and to cool the second floor in the evening. In new installations the duct systems could be so designed that only one or two dampers would require changing in

order to divert the cooling effect from one floor to the other.

The arrangement of the ice tank was quite satisfactory. The standby loss was about 1,600 Btu per hour or 10 lb of ice. It was found desirable to use only a small amount of water in the bottom of the tank because there is an appreciable irrecoverable loss if a large volume of water is cooled. An inside tank could be used but should be provided with a charging chute from outside and should be insulated against noise and excessive heat transfer.

The matter of ice delivery will demand study on the part of the ice companies when this method of cooling becomes more common. Deliveries must be prompt because it is sometimes difficult to predict more than three or four hours in advance when ice will be needed. Some method of scheduling deliveries according to the outdoor temperature may be desirable.

Conclusions

1. The results of this investigation lead to the definite conclusion that in a well-shaded and insulated house, in the Detroit, Michigan climate, summer cooling with ice is economically feasible provided a moderate amount of cooling is done.
2. A well designed mechanical warm air heating system is readily adapted to summer cooling and the additional equipment required for cooling is not extremely costly.
3. The use of outside air at such times as its temperature is below the inside temperature is desirable but does not, by itself, give adequate comfort in extremely hot weather.

Local Chapter Reports

Cleveland

December 9, 1932. When 25 members and guests of the Cleveland Chapter had assembled in the Chapter Rooms of the *Cleveland Engineering Society*, the meeting was called to order and Secy. F. A. Kitchen reported on the recent ruling of the Council in regard to dues and initiation fees. Pres. F. B. Rowley's letter to Society members was read and the Annual Meeting in Cincinnati was called to the attention of those present.

J. C. Miles, chairman of the Program Committee, then introduced the speaker of the evening, A. J. Nesbitt, of Philadelphia, whose subject was A Logical Basis for Determining the Air Volume to Be Circulated in Classroom Ventilation. Mr. Nesbitt gave a very interesting talk covering the experiments and tests he had made in Philadelphia schools and illustrated his findings by means of slides. The members present entered into a lively discussion and by a rising vote of thanks to Mr. Nesbitt indicated that they had been present at one of the most interesting meetings of the year.

Kansas City

January 9, 1933. At the January meeting, held in the Newburn Hotel, the subject of reduction in governmental costs was discussed by Walter Matscheck, Director of the Kansas City Civic Research Institute. Mr. Matscheck presented a most able study of the various factors which enter into any reorganization of government.

The regular educational feature was a discussion of the Society's Ventilation Standards by N. W. Downes and his presentation was followed by a lively discussion, which gave the members present a better understanding of the problems involved.

Plans were made for the Joint Meeting in February at which an attendance of 500 people is anticipated.

The Annual Meeting in Cincinnati was discussed and members were urged to be present. Benjamin Natkin will represent the Kansas City Chapter on the Nominating Committee and T. L. Dawson was named as alternate.

The Membership Committee showed that promising results were being obtained on the new basis of reduced dues.

December 12, 1932. The members and guests enjoyed a well-attended meeting at the Newburn Hotel, at which the guest speaker was M. A. Smith, of Chicago. Mr. Smith gave an interesting and entertaining discussion on Various Phases of Acoustical Treatment of Rooms and Machinery to Eliminate Objectionable Noise.

A paper on Air Movement was given by E. K. Campbell and was based on his experience on unusual installations. His talk proved to be most interesting and brought forth a very lively discussion.

Arrangements were also planned for the combined meeting of the *A. I. E. E.*, the *A. S. M. E.* and the A. S. H. V. E. to be held in February.

Illinois

December 12, 1932. Vice-Pres. C. W. DeLand opened the meeting in the absence of Pres. J. H. O'Brien and welcomed the guests of the Illinois Chapter who had assembled in the Hotel Sherman.

The members were informed of the death of August Kehm, one of their most beloved members, and a moment of silent tribute followed John Hale's description of Mr. Kehm's early relationship with the Chapter and his unfailing interest and support.

G. H. Blanding instituted the feature, Ten Minutes of Something New, and spoke on Relative Humidity and Some New Developments in Recording Instruments.

W. H. Driscoll, New York, was introduced by A. B. Martin as the principal speaker, and his subject, A Romance of Building, told of the construction of the Waldorf Astoria Hotel. Mr. Driscoll prefaced his talk with a few timely remarks and urged new members and old to carry on in these difficult times for the good of the Society. He then presented a vivid picture of the reclaiming of the Park Ave.-Grand Central Terminal District in New York, culminating in the erection of the Waldorf Astoria, and qualified as an artist as well as an engineer by his descriptions.

An enthusiastic vote of thanks was given to Mr. Driscoll for his splendid talk.

Massachusetts

January 9, 1933. Prior to the regular business meeting of the Massachusetts Chapter, 27 members gathered for luncheon in Hopkins Restaurant. Pres. F. R. Ellis announced that E. Q. Cole had been designated as representative and W. W. Murphy as alternate on the Nominating Committee of the Society.

The meeting was then turned over to J. S. Webb, of the Papers and Meetings Committee, who spoke briefly and recounted some of the experiences of others in securing business during these times. As previously stated, Mr. Webb concluded his remarks just one hour after the time of assembly. A short discussion ensued and the meeting was adjourned at 2:00 o'clock.

December 12, 1932. A luncheon meeting at the Engineers Club in Boston was enjoyed by 40 members and guests of the Massa-

chusetts Chapter. A letter written by Pres. F. R. Ellis to the *Engineering Societies of Boston* regarding the listing of engineers was read, as was the reply which had been received.

At the close of the business meeting, Paul L. Sackett, gave a paper entitled, Some Recent Air Conditioning Applications. Mr. Sackett described several of the installations which had recently been made and told of the reasons for the various treatments used.

Following a discussion, the meeting was adjourned at 2:00 p. m.

Michigan

December 12, 1932. Thirty members and guests attended a dinner meeting of the Michigan Chapter, at which Pres. L. L. McConachie asked for opinions on the Summer Meeting 1933. The various phases of a boat trip as compared to a meeting held in the city were discussed.

The meeting was then turned over to L. L. Smith, who introduced E. P. Wells. Mr. Wells spoke on Building Trades Relations from Building Owners Viewpoint.

Many helpful suggestions were offered giving ways of increasing the attendance at meetings.

Western Michigan

December 12, 1932. The value of the technical meetings of the Western Michigan Chapter was emphasized when, in addition to local residents, 20 members and guests traveled considerable distances in very severe weather to attend the meeting in Grand Rapids.

A. J. Nesbitt, Philadelphia, Pa., presented a paper on A Logical Basis for Determining the Volume of Air for Classrooms and, with the aid of stereopticon slides, described a series of experiments conducted in a Philadelphia school, in which an endeavor was made to find an optimum of ventilating conditions for the room occupants, while reducing operating costs to a minimum.

An extended discussion followed and the Chapter is very appreciative of Mr. Nesbitt's kindness in making a special trip from the East.

Pres. D. L. Taze announced the appointment of S. H. Downs as representative and J. H. Van Alsburg as alternate on the Society's Nominating Committee.

The adjournment marked the close of another enjoyable and instructive meeting.

New York

January 16, 1933. The January meeting of the New York Chapter consisted of a Symposium on Refrigeration which was held at the Building Trades Club, New York, and was attended by 250.

Dinner was served at 7:00 p. m., and after a short business meeting a token of appreciation was presented by the Chapter to F. E. W. Beebe for his 10 years of faithful service as treasurer of the Chapter. The presentation was made by National Past-President W. H. Driscoll, who very highly praised the fine work of Mr. Beebe in serving the Chapter.

Four speakers were included on the program, namely, W. L. Fleisher, Consulting Engineer, New York, an expert on refrigeration and air conditioning; Claude A. Bulkeley, chief engineer of the Niagara Blower Co.; R. W. Waterfill of the Carrier Corporation, and a member of the *American Society of Refrigerating Engineers*; C. F. Holske, test engineer of the American Ice Co.; and E. W. Legier of the American Blower Co.

Mr. Fleisher delivered a very interesting talk on the general phases of air conditioning and called attention to many points with which the average engineer is not familiar.

Mr. Waterfill discussed the Steam Jet Refrigeration and the Centrifugal Refrigerating Systems. His talk was illustrated by slides.

Mr. Bulkeley discussed some fundamental points in connection with utilization of air and water in the air conditioning process.

Mr. Holske delivered a very interesting and instructive talk on the use of ice for cooling and air conditioning processes, while Mr. Legier confined his talk to the presentation of interesting facts relative to unit air conditioning.

Each of the speakers was allowed 30 minutes, after which a lively open discussion took place.

December 19, 1932. An interesting meeting of the New York Chapter was held at the Building Trades Club when 60 members and guests assembled to hear Mr. VanderVeer, a member of Commander Byrd's expedition to both the North and South Poles.

Pres. L. T. M. Ralston announced the death of a well known member, George D. Hoffman, and stated that the sentiments of the Chapter would be expressed to his family.

Samuel R. Lewis, Chicago, Past President of the Society, was a guest and was introduced to the meeting.

A large attendance at the Annual Meeting of the Society in Cincinnati was urged and transportation arrangements for Chapter members were outlined.

In accordance with a custom established several years ago, H. Berkley Hedges presented the past president's trophy to Russell Donnelly, formerly President of the chapter. Mr. Donnelly responded and expressed his appreciation to the members.

Mr. VanderVeer was then introduced and presented sound motion pictures of his travels in the Antarctic. It was learned that he had been a member of expeditions to both the North and South Poles, the tropics, and that his experiences included being shipwrecked and arrested as a spy.

The meeting was thoroughly enjoyed by those present and adjourned at 11:00 p. m.

Western New York

January 9, 1933. The annual election of officers was held at the January meeting of the Western New York Chapter and the following were unanimously selected: President—D. J. Mahoney; 1st Vice-President—J. J. Yager; 2nd Vice-President—W. F. Johnson; Secretary—W. E. Voisinet; and Treasurer—William Roebuck, Jr.

The members then were privileged to hear T. A. Novotney, Johnstown, Pa., who spoke on Convection Heaters. Mr. Novotney discussed heaters particularly from a research viewpoint.

December 19, 1932. This meeting was the Annual Christmas Stag Party of the Chapter and was held at the University Club through the courtesy of D. J. Mahoney. It was well attended and everyone enjoyed the bowling, cards and buffet supper.

A short business meeting was called by Pres. M. C. Beman, who presented a telegram from the Nominating Committee containing the slate of officers to be voted on in January.

Many other telegrams of a facetious nature were read and were enthusiastically received with great applause.

Minnesota

December 12, 1932. The regular meeting of the Minnesota Chapter was held at the Men's Union Building, University of Minnesota, with 31 present. At the conclusion of dinner, Pres. W. F. Uhl announced that the time was to be devoted to a series of papers on elementary and basic data for the design of air conditioning systems and equipment.

Albert Buenger, who had arranged the program, took charge of the meeting and C. E. Lewis presented a paper on Fundamental Air Conditions as Required by the Human Body. Mr. Lewis explained the actual operation within the body itself of temperature control and the part played by respiration and metabolism.

Other papers were given by A. B. Algren on The Essential Requirements of an Air Conditioning System; E. F. Jones on The Various Definitions of Temperatures; D. M. Forfar on The Definitions of Humidities; and M. S. Wunderlich on Coefficients Used in Figuring Heat Transmission Losses. The entire

membership joined in a general discussion of the various papers and their relationship to one another.

The program was received enthusiastically and it was voted to hold a like meeting in January, starting at 4:00 p. m. in the Experimental Engineering Building at the University of Minnesota.

Philadelphia

January 12, 1933. The Philadelphia Chapter held its annual meeting at the Engineers Club and dinner was served in the auditorium before Pres. L. C. Davidson called the meeting to order.

A. C. Caldwell, chairman of the Auditing Committee, submitted the committee's report and the audit was accepted.

President Davidson read extracts from a communication received from the secretary of the Philadelphia Technical Service Committee, expressing thanks for the work of the Chapter in raising funds for the committee.

The Annual Meeting of the Society in Cincinnati was announced by F. D. Mensing and attention was called to the reduced railroad rates.

A humorous specification which was drawn up at the time of the organization of the Philadelphia Chapter was read by J. D. Cassell.

The list of nominations for Chapter officers and Board of Governors was presented by President Davidson and the following were elected to serve in 1933: *President*—M. F. Blankin; *Vice-President*—W. P. Culbert; *Secretary*—W. R. Eichberg; *Treasurer*—J. H. Hucker; and *Board of Governors*—L. C. Davidson, W. F. Smith and A. McClintock, Jr.

Mr. Davidson was presented with a very handsome golf bag from the Chapter as an expression of their appreciation of his efforts during his term of office.

The speaker of the evening was W. K. Simpson, Waterbury, Conn., who gave a very interesting paper on Problems in Design of Thermostatic Traps, illustrating his talk by slides.

December 8, 1932. A dinner and meeting of the Philadelphia Chapter was held in the Engineers Club and, prior to the regular business, S. M. Heilborn gave a short talk on the United Campaign.

F. D. Mensing briefly presented interesting facts in connection with the Society's Annual Meeting in Cincinnati.

Pres. L. C. Davidson called on H. G. Black for a report of his committee which was assisting the Philadelphia Technical Service Committee. Mr. Black stated that collections slightly exceeded the Chapter quota, which was in the amount of $1,250.00 and also said that the Chapter was the only organization associated with the Engineers Club that had reached its allotment.

Instead of making the usual January meeting one of entertainment, it was decided to hold a regular meeting at the Engineers Club and to distribute the amount usually spent to assist members of the Chapter who are in distress.

A. J. Nesbitt, a Chapter member, then presented a paper on A Logical Basis for Determining the Volume of Air in Classroom Ventilation and held the interest of everyone present.

Pittsburgh

December 12, 1932. The annual meeting of the Pittsburgh Chapter was held in the Fort Pitt Hotel and Pres. R. B. Stanger called the assembly to order.

An appeal was made to the members to pay their dues promptly and the secretary, J. L. Blackshaw, read a report on membership status and a letter from R. H. Carpenter, chairman of the Committee on Increase in Membership of the Society, which outlined the new basis of membership dues and initiation fees.

President Stanger presented the nominations made by the Nominating Committee and the secretary was instructed to cast a unanimous ballot for the slate as presented. The following officers were elected: *President*—G. S. McEllroy; *Vice-President*—P. A. Edwards; *Secretary*—J. L. Blackshaw; *Treasurer*—H. B. Orr; *Board of Governors*—R. B. Stanger, Peter O'Neill and F. H. Hecht.

Mr. Stanger continued as chairman and E. C. Evans reviewed the new proposed Constitution and By-Laws of the Society. The meeting then took the form of a smoker and informal round-table discussion of Society affairs.

H. Lee Moore spoke of A. S. H. V. E. activities, tracing the progress of the Society and citing especially the Code of Ethics. Secretary Blackshaw gave an illustrated talk on activities of the Research Laboratory.

The work of the Research Laboratory was praised highly by V. W. Hunter, who read a paper showing the economies effected as a result of using the facts developed through research on garage ventilation methods. After a discussion of his talk, Mr. Hunter was requested to prepare a testimonial for presentation to the Committee on Research and the Council of the Society.

The next presentation was by Prof. C. M. Humphreys, of Carnegie Institute of Technology, who spoke of the value of the Society to the student and especially emphasized the value of THE GUIDE.

St. Louis

January 4, 1933. Pres. C. A. Pickett called the St. Louis Chaper Meeting to order in the Roosevelt Hotel with 30 members and guests present.

The Chapter approved the action of the Board of Governors reducing annual chapter dues and continuing the present status of Limited Chapter Members.

Secretary C. R. Davis read a message from Pres. F. B. Rowley to A. S. H. V. E. Members with reference to reduced dues and initiation fees, the Annual Meeting, and the Revision of the Constitution and By-Laws.

President Pickett called for reports from the chairmen of all committees and L. W. Moon, of the Program Committee, stated that an invitation had been received to meet in February with the heating and ventilating class of the David Ranken Trade School and a vote decided in favor of this action. Other chairmen reporting were J. W. Cooper for the Entertainment Committee, Paul Sodemann, Membership Committee, G. W. F. Myers, Publicity Committee, and E. B. Evleth of the Auditing Committee.

Paul Sodemann was then introduced by Mr. Moon and presented a very interesting and instructive paper on Conditioning Air by the Application of Refrigeration. In his talk Mr. Sodemann covered the working out of a space cooling problem and outlined the method of calculating different losses. He also described the methods now in use for applying refrigeration in air conditioning work and the characteristics of different refrigerants. A rising vote of thanks was given to Mr. Sodemann for one of the best papers presented in some time.

Wisconsin

December 14, 1932. A short business session was held at which the relative merits of a luncheon meeting were discussed from the standpoint of attendance and of securing a satisfactory speaker. This suggestion was referred to the Program Committee for consideration and further action.

E. A. Jones was appointed as the Chapter's representative on the Nominating Committee with Ernest Szekely as alternate.

The guests attending the dinner were introduced by Pres. V. A. Berghoefer, after which Mr. Szekely presented the speaker, A. J. Nesbitt, of Philadelphia.

Mr. Nesbitt gave his paper on A Logical Basis for Determining the Volume of Air to be Circulated in Classroom Ventilation and his talk proved to be one of the most interesting. Considerable discussion followed before the meeting was adjourned.

A. M. Byers Co. Makes Appointment

M. J. Czarniecki, formerly Manager of Tubular Sales of A. M. Byers Co., has been appointed General Manager of Sales. In

announcing the promotion, H. W. Rinearson, Vice-President, stated that the move had been necessitated by the departmentalizing of sales activities due to the increased variety of wrought iron products developed by the company within the past year.

L. L. Munier President of Wolff & Munier

With the retiring of R. A. Wolff from active participation in Wolff & Munier, Inc., and his resignation as president, L. L. Munier was elected to succeed him.

Mr. Wolff will continue, however, as a member of the Board of Directors.

A. S. A. Elects New Officers

Howard Coonley, president of the Walworth Co., New York, N. Y., has been elected to the presidency of the *American Standards Association* for 1933, following several years of active connection with the standardization movement as representative of the *American Society of Mechanical Engineers.*

F. E. Moskovics, chairman of the Board of Directors of the Marmon-Herrington Co., Indianapolis, Ind., is now vice-president of the *A. S. A.* and previously represented the *Society of Automotive Engineers.*

Associated in Engineering Practice

Josiah H. Smith, formerly chief engineer of Ballinger Co., has become associated with Wm. G. R. Braemer in the practice of engineering with offices at Room 1265, Commercial Trust Bldg., Philadelphia, Pa. They are prepared to render a comprehensive engineering service, which includes design of industrial, power, sewage disposal, refrigerating, air conditioning, heating and ventilating plants, also comfort cooling, humidifying, dehumidifying, automatic temperature and humidity regulation.

W. G. Kreitner Joins Building Expense Control Co.

William G. Kreitner has been appointed vice-president in charge of heating and ventilating of the Building Expense Control Co., Inc., New York, N. Y., building managers of numerous office and apartment buildings.

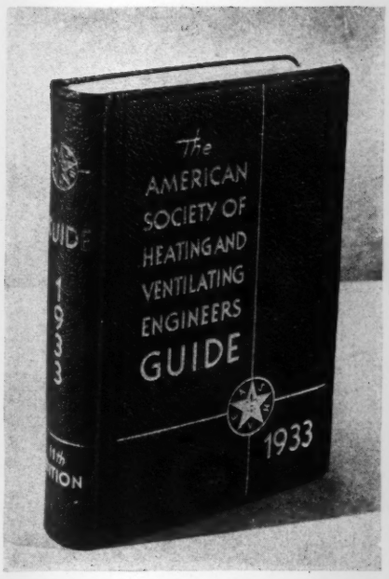
Mr. Kreitner was formerly heating engineer with Modine Mfg. Co. and engineer in charge of construction and service for the Milwaukee Valve Co.

New Edition of Refrigeration

The second edition of Refrigeration by Moyer and Fittz has recently been announced by the McGraw-Hill Book Company, New York.

Progress in the fields of refrigeration and air conditioning during recent years has made necessary a complete revision of this book. For the reason that the previous edition has been extensively used as a textbook in engineering and agricultural schools, the chapters that deal with the thermodynamic theory of refrigeration have been extended. The needs of practical engineers have, however, been constantly in mind during the preparation of the revision so that the usefulness of the book for reference purposes has not been abridged. The index that has been an important factor in making this book adaptable for ready reference has been improved and extended adequately to cover the subjects that have been added.

Important new features, such as improved designs of household and small commercial refrigerating units, silica-gel refrigerating system, new small-capacity absorption plants, methods of controlling refrigerants, revised data on the properties of refrigerants, production of solid carbon dioxide (dry ice), applications of quick freezing and air conditioning, as well as many improvements in refrigeration services, have made necessary the enlargement of the chapters of the preceding edition and the addition of new ones.



The New Guide is Here

THE GUIDE 1933 is just off the press. This 11th Annual Edition of the standard reference volume on heating, ventilating and air conditioning appears in an entirely new "dress" and the contents have been extensively enlarged and revised to include the latest results of research and modern engineering practice. Compiled by the foremost engineers in the profession, THE GUIDE 1933 embodies in its 45 chapters not only the data developed at the Society's Research Laboratory and cooperating institutions, but also the most practical and useful ideas of outstanding engineers in the profession.

The Text Section of THE GUIDE 1933 contains 608 pages, supplemented by 180 pages of Manufacturers Catalog Data with an Index to Modern Equipment, also 64 pages of the Society's Roll of Membership.

Eleven new chapters are to be found and extensive changes are to be noted in the other sections which have been retained. Chapter 3, dealing with Transmission Losses, presents an entirely new set-up of tables with coefficients of transmission based upon the latest investigations of the Society's Research Laboratory in Pittsburgh and the most recent data obtained at the Experimental Engineering Laboratories of the University of Minnesota. Chapter 14 gives additional information on Heating Boilers.

The Society's new Ventilation Standards, adopted in 1932, are included in THE GUIDE with other important data in Chapter 22, relating to Ventilation and Air Conditioning for Comfort and Health. Much of the information came from cooperative studies at the School of Public Health, Harvard University. Considerable information heretofore unpublished will be found in Chapters 24 and 25 and the data on Air Duct Design (Chapter 33) has been amplified and made more useful by the introduction of more complete examples, showing the methods of determining accurate duct sizes. The section describing Test Instruments and Methods (Chapter 39) is new material. Chapter 36 is devoted to Natural Ventilation and presents new data. Among the chapters, which have had extensive revision are Chapter 4, Infiltration Heat Losses; Chapter 6, Radiators and Gravity Convectors; Chapters 9 and 10 dealing with Steam Heating Systems; Chapters 16, 17 and 18 dealing with Fuels and Combustion; Chapter 38, Fans and Motive Power; and Chapter 43 on Smoke and Dust Abatement.

THE GUIDE 1933 depicts with startling clarity the scope of developments that have occurred in the field of heating, ventilating and air conditioning within a short period of time. The value of THE GUIDE, because of the authoritative character of its text, has been recognized by thousands of engineers, architects